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### About Authors

• **E. E. Aldrin** *was the first American to make a business trip through Europe by airplane. He is president of the Daniel Guggenheim Medal Fund and vice-president of the Institute of Aeronautical Sciences. Prior to his business career, Major Aldrin was a regular in the U. S. Army from 1917 to 1928. He got his Air Service commission in 1920. When the Italian squadron flew the Atlantic to America in 1934, Major Aldrin rendered them such valuable assistance that he later was given the decoration of Commendatore of the Order of the Crown of Italy. He has a Doctor's degree in engineering from M.I.T. and did his undergraduate work at Clark University and Worcester Polytechnic Institute.*

• **Paul A. Anderson** *has followed aircraft engineering ever since he left school in 1918. His efforts began at McCook Field. Ten years were devoted to aircraft engineering at several aircraft factories, and the past eight years to aircraft-engine testing and installation at Wright Aeronautical Corp. as designer, project engineer and field engineer. Mr. Anderson was born in Indiana.*

• **E. M. Barber** *was the Elliott Fellow at Penn State's engineering experimental station before he joined the staff of Texas Co.'s Beacon laboratory in 1933. He has been engaged in lubrication and fuel research projects ever since getting his B.S. at Penn State in 1931. He was born in Oil City, Pa.*

• **H. E. Buc** *came to America at the age of 14, from his birthplace, Russia. Since 1920 he has worked on various petroleum problems for the Standard Oil Development Co. With the Bureau of Chemistry from 1911 to 1919, his specialty was alkaloids. He was graduated from College of the City of New York in 1901 and studied at Columbia from 1908 to 1911.*

• **George Hook** *first was inoculated with the highway transportation virus while editor of an English language newspaper in Havana. The island was badly in need of good roads, so Hook ran a daily front-page interview with visitors from U. S. A. on the*

*(Continued on page 24)*

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# *Greyhound Chooses* ROSS *for* STEERING STABILITY

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CLEVELAND, OHIO

July 15, 1936.

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# ROSS CAM & LEVER STEERING

# A New High-Octane Blending Agent

By H. E. Buc and Major Edwin E. Aldrin
*Standard Oil Development Co.*

**TO describe a new high-antiknock fuel that is well suited for the modern high-output aircraft engine and potentially available in large quantities, is the object of this paper. The high-octane blending agent used is isopropyl ether.**

**Results of full-scale multicylinder engine tests with this material and iso-octane each blended with aviation gasoline to give a 100-octane fuel with 3 cc. of tetraethyl lead, and with iso-octane blended with aviation gasoline to give a 92-octane fuel with 3 cc. of tetraethyl lead, indicate that:**

**Minimum specific fuel consumption of the isopropyl-ether 100-octane blend is lower under cruising conditions than the 92-octane, but higher than the 100-octane, iso-octane blend; the lower economy of isopropyl ether may be overcome by going above 100-octane number; and the 100-octane blends of isopropyl ether and iso-octane are equal in power output and consumption under high-power conditions.**

**The material easily meets water-tolerance specifications showing only slight solubility in water. Low gum content and satisfactory resistance to oxidation of these blends indicate the storage stability proved by storage tests.**

**Single-cylinder engine tests show that isopropyl ether has a blending value slightly superior to technical iso-octane and less lead is necessary to meet a given octane number for equivalent percentage concentrations.**

**A recent survey of the supply of propylene, the raw material, indicates enough of it available to produce 850,000,000 gal. per year of 100-octane fuel – more than sufficient even in a National emergency.**

IN the modern aircraft engine the increased power output for a given piston displacement has been made possible mainly by improvement in fuels to permit higher compression and greater supercharger boost. In 1930 the U. S. Army Air Corps used a fuel ranging between 75- and 85-octane number (A.S.T.M.) and, in 1931, raised this standard to about 84-octane number and later to 87-octane number (A.S.T.M.). Since 87-octane fuel has been available for the past five years and since the possibilities of higher octane fuels in permitting greater advances in engine design, output, and economy are recognized generally, the obvious question arises as to why the antiknock value of aviation fuels has taken no definite upward rise during this period.

It must be recognized that the attainment of high-octane number in a natural fuel, and by high octane is meant a value approaching or exceeding that of pure iso-octane, is not only a scientific but also an economic problem, considering that sufficient quantities must be in sight for any possible requirement before engines for its proper and exclusive utilization will be used generally. Within limits high-octane fuels have been available and used experimentally to demonstrate the advantages of high-output engines through the medium of using higher than 3 cc. of tetraethyl lead per gal. and, more recently, through the medium of technical iso-octane which is now available in commercial quantities at a cost which, in many cases, makes its use profitable.

The object of this paper is to describe a new fuel of high antiknock value apparently suitable in all respects for the modern high-output aircraft engine and potentially available in large quantities.

There still exists considerable confusion in regard to evaluating the antiknock value of aviation fuels in terms of flight performance as related to any single-cylinder engine test. The C.F.R. Aviation Gasoline Detonation Subcommittee is working diligently on this problem, and an early acceptable solution is to be desired. In the meantime, many airlines and aircraft-engine manufacturers are using A.S.T.M. ratings in specifications; the British Air Ministry in collaboration with the I.P.T. has adopted a slightly modified A.S.T.M. rating to suit British aircraft engines; and in this country the U. S.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 5, 1936.]

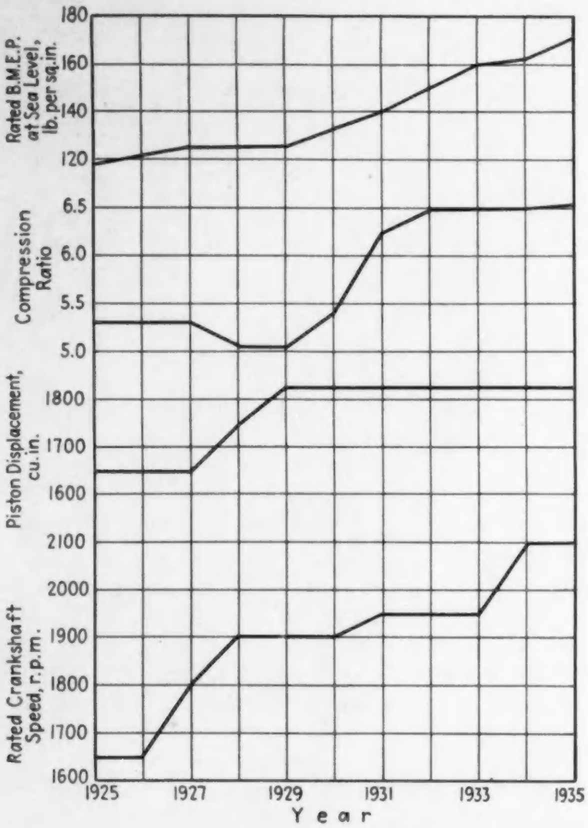



**Fig. 1 – Cyclone Engine Trends Over a Ten-Year Period**

Army Air Corps has been most active in research activities on high-output engines and high-octane fuel tests and has developed its own single-cylinder test for such fuels. This test involves the use of a modified C.F.R. engine, and a large amount of flight experience appears to have justified its use pending further improvements. In the remaining parts of this paper, octane number will be understood to be by the Army test method, unless otherwise designated.

Developments in engine design during the past ten years are indicated roughly in Fig. 1, taken from a paper[1] presented at the Annual Meeting of the Society, Detroit, Jan. 17, 1936, by Raymond W. Young of the Wright Aeronautical Corp. Fig. 2 from the same paper shows the variation in cylinder-head temperature for different specific fuel consumptions when using various octane-number fuels (A.S.T.M.) in a 1930 Cyclone 600-hp. engine and a 1935 Cyclone 800-hp. engine. The latter curve, indicating the minimum specific consumption possible without either misfiring or a sharp rise in cylinder-head temperature, illustrates a significant and important benefit obtainable with high-octane fuels.

When engines are designed to cruise at best economy on a given fuel, serious limitations are encountered at full-power take-off, during climbing, and in emergency conditions. These limitations are overcome in practice either by providing a special fuel of higher octane number than required for cruising, or by operating at rich mixtures when delivering full power. Neither is a satisfactory solution, the first because of obvious reasons necessitating dual tankage, fuel controls, and a personal factor in pilot conduct; the second for reasons of economy. For maximum cruising economy it would be desirable to use the same fuel as employed for take-off.

[1] See S.A.E. TRANSACTIONS, June, 1936, pp. 234-256; "Air-Cooled Radial Aircraft-Engine Performance Possibilities", by Raymond W. Young.

The experience of airlines and engine manufacturers has shown that, with present design and existing materials, a moderate concentration of tetraethyl lead is desirable in controlling and stabilizing combustion in high-output engines and, at the present time, the best blended fuel would appear to be one of approximately 100 Army octane number containing 3 cc. of lead per gal.

## Introduction to Main Subject

Petroleum chemists the world over have for many years been seeking new materials suitable either as high-antiknock fuels by themselves or as blending agents in the preparation of base stocks to which some antiknock compound, such as tetraethyl lead, could be added in small quantities to give the desired high antiknock value. Technical iso-octane is one such material already made commercially available.

The petroleum industry has now succeeded in providing a new material to supplement technical iso-octane. This material is isopropyl ether. Contrary to the behavior of ethyl ether, which is a violent knock-inducer, isopropyl ether, along with some other ethers, has been found to be an exceptionally good antiknock fuel. During the past eight years, experimentation has progressed on this material and methods of manufacture studied until at this time it appears to be an important factor in the successful solution of the high-antiknock-fuel problem. Full-scale multicylinder-engine tests, mentioned later, have demonstrated its remarkable antiknock qualities and, in physical characteristics, it apparently meets the requirements of present-day military and commercial aviation-fuel specifications when blended with aviation gasoline in proportions to give 100 octane with 3 cc. of lead tetraethyl per gal.

The physical characteristics of isopropyl ether are shown in Table 1, along with corresponding values for pure iso-octane and benzene shown for comparison. Although a low-

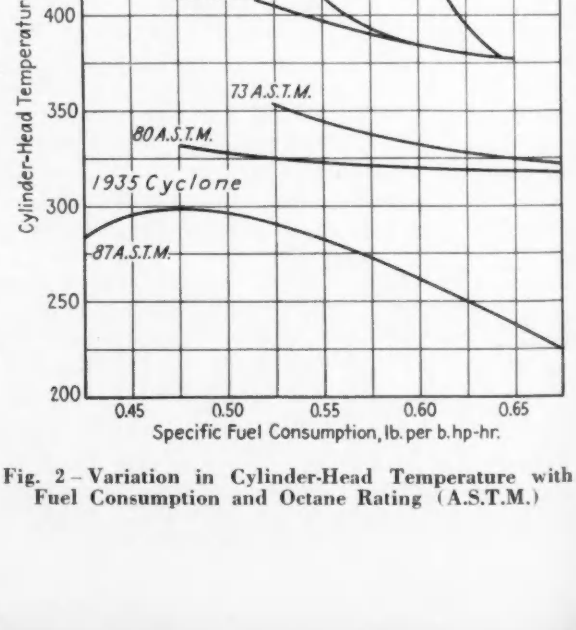



**Fig. 2 – Variation in Cylinder-Head Temperature with Fuel Consumption and Octane Rating (A.S.T.M.)**

**Table 1 – Physical Characteristics of Pure Compounds**

| | Isopropyl Ether | Pure Iso-Octane | Benzene |
|---|---|---|---|
| Boiling Point, deg. fahr. at 760 mm. hg. pressure | 154 (153-158) | 211 | 175 |
| Density at 68 deg. fahr. (20 deg. cent./4 deg. cent.) | 0.725 | 0.691 | 0.878 |
| Refractive Index at 68 deg. fahr. | 1.3680 | 1.3921 | 1.5014 |
| Freezing Point, deg. fahr. | −125† | −162 | 42 |
| Viscosity, centipoise | 0.322 (68 deg. fahr.) | 0.543 (65 deg. fahr.) | 0.647 (68 deg. fahr.) |
| Latent Heat of Vaporization, B.t.u. per lb. | 123 | 130 | 170 |
| High Value of Heat of Combustion, B.t.u. per lb. | 16,900 | 20,580 | 17,650 |
| Low Value of Heat of Combustion, B.t.u. per lb. | 15,600 | 19,200 | 17,000 |
| Low Value of Heat of Combustion plus Latent Heat of Vaporization, B.t.u. per gal. | 95,100 | 111,400 | 125,700 |
| Reid Vapor Pressure, lb. per sq. in. at 100 deg. fahr.* | 5.3 | 2.2 | 3.2 |

*Reid vapor pressure was determined on commercial products rather than on c.p. compounds. In case of benzene, vapor-pressure value is for commercial 90 per cent benzol.
†Particular emphasis should be placed on the freezing point of isopropyl ether. Freezing which has occurred with benzol in practice may have been due to benzol itself. Some freezing has occurred at very high altitudes and low temperatures in exposed fuel systems from water obtained in ordinary gasoline. Also, attention is called to the latent heat of vaporization for isopropyl ether, it being even better than gasoline from the carburetor-refrigeration standpoint.

boiling-point material, its vapor pressure is below the maximum allowed for aviation gasolines and, when blended therewith, vapor pressures will be entirely suitable. Freezing point is low and thus it does not have the disadvantage of the high freezing point of the benzene or benzol-blend fuels. Latent heat of vaporization, while lower than for benzene, is comparable with that for iso-octane and, consequently, the same maximum engine horsepower would be anticipated from considerations of mixture-cooling effect, and no increase in the tendency to cause carburetor icing due to excess refrigerating effect would be anticipated. Heating value on both a weight and volume basis is lower than for either iso-octane or benzene, and this property normally would be expected to give a reduction in miles per gallon or, conversely, would require greater tank capacity for a given cruising radius. The magnitude of this disadvantage is in proportion to the quantity of isopropyl ether used in the final blend and, as will be shown later, it may be wiped out in practical blends by permitting of leaner air-fuel ratios without excessive head temperatures.

In considering the physical characteristics of aviation gasolines, the subject of water tolerance is of major importance. The existing specifications for water tolerance of aviation fuels require only that, when 80 ml. of the fuel is shaken with 20 ml. of water at room temperature, the increase in volume of the water layer shall not exceed 2 ml. This requirement is directed against the use of alcohol blends from which the alcohol usually is separated easily by the addition of water. Isopropyl ether has only very slight solubility in water, and the preceding water-tolerance test usually shows no measurable increase in the water layer, even when concentrations of 40 per cent of this material are used in blends with aviation gasoline.

The low solubility of isopropyl ether in water was demonstrated by a more accurate analysis than the previously described rough test. 503 ml. of a 40 per cent blend of isopropyl ether in aviation gasoline was shaken with 497 ml. of water until equilibrium was established. The final contraction of the gasoline layer was 2.63 ml. or 0.52 per cent. The water content of the gasoline layer was then determined (by a method described later) as 0.09 per cent. Since no change was found in the total volume (1000 ml.) of gasoline and water, the total loss of isopropyl ether by solution in water was 0.61 per cent on the basis of the original blend or 1.5 per cent of the iso-propyl ether itself.

The preceding results indicate the magnitude of the loss of fuel that might possibly be suffered if a blend of isopropyl ether were stored over water for long periods of time. This possibility has practical significance in view of the occasional use, for example in the U. S. Army Air Corps, of water-displacement storage systems.

However, another aspect of the problem of water tolerance, which is sometimes of importance, is the tendency for aviation gasoline to dissolve water which later may be deposited at freezing temperatures as ice in the fuel-feed systems of airplanes in flight. This difficulty has been encountered occasionally in the past with aromatic blends.

In this connection, the most practical consideration is not the amount of water dissolved in the gasoline at any given temperature, but rather it is the amount of water that drops out of solution as ice at any given temperature below the freezing point of water. Accordingly, in comparing the water tolerance of isopropyl ether with that of other types of aviation fuel, two different methods of determination were used. In one set of tests, the total water content of the fuel, when saturated with water at 77 deg. fahr. or at 32 deg. fahr., was determined. In these tests, calcium hydride was added to the water-saturated fuels, and the hydrogen evolved by reaction with the water was measured accurately in a gas burette and calculated back to its equivalence in ml. of water per 100 ml. of fuel. The water contents of three types of fuel were as follows:

| | Ml. of Water per 100 Ml. of Fuel Saturated at 77 deg. fahr. | Ml. of Water per 100 Ml. of Fuel Saturated at 32 deg. fahr. |
|---|---|---|
| 100 Per Cent Regular Aviation Gasoline | 0.007 | 0.006 |
| Aviation Gasoline with 40 Per Cent Benzol | 0.022 | 0.019 |
| Aviation Gasoline with 40 Per Cent Isopropyl Ether | 0.085 | 0.062 |

In the second series of tests in which the actual water separation was measured, it was shown that, although the isopropyl-ether blends dissolve more water at 77 deg. fahr., they also retain more water in solution at temperatures as low as −20 deg. fahr. Consequently, the tendency to separate is not as great as it appears from the water content at 77 deg. fahr.

In the second series of tests, the ice separation at −20 deg. fahr. was determined directly. The aviation fuels were saturated with water at 70 deg. fahr. and then pumped first through a precooler at 32 deg. fahr. and thence into an ice trap at −20 deg. fahr. At the end of each test, the ice trap was disconnected and its water contents were distilled with

**Table 2 – Storage Tests on Aviation Gasoline in Hot Room at 100 Deg. Fahr.**

Blend: 60 Per Cent Aviation Gasoline, 40 Per Cent Isopropyl Ether, Unleaded and with 3 Cc. Lead per Gal., and with 1.4 Mg. B.A.P. Inhibitor per 100 Ml.

| Blend | 50 Gal. Unleaded | | | | 50 Gal. Leaded | | | | 5 Gal. Unleaded | | | | 5 Gal. Leaded | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Storage Container | New Galvanized Drum | | Used Galvanized Drum | | New Galvanized Drum | | Used Galvanized Drum | | Tin Can | | Tin Can + 1 Qt. Sea Water | | Tin Can | | Tin Can + 1 Qt. Sea Water | |
| Date of Sample, Start—1/30/36 | 1/30 | 4/13 | 1/30 | 4/13 | 1/30 | 4/13 | 1/30 | 4/13 | 1/30 | 4/13 | 1/30 | 4/13 | 1/30 | 4/13 | 1/30 | 4/13 |
| Copper-Dish Gum (mg. per 100 cc.) | 4.5 | 4.0 | 4.5 | 3.0 | 4.5 | 4.0 | 4.5 | 3.0 | 3.5 | 2.5 | 3.5 | 3.5 | 3.5 | 4.5 | 3.5 | 4.0 |
| Army Gum (mg. per 100 cc.) | 1 | 2 | 1 | 2 | 1 | 2 | 1 | Trace | 1 | 1 | 1 | Trace | 1 | Trace | 1 | 1 |
| Air-Jet Gum (mg. per 100 cc.) | 1 | 1 | 1 | 1 | 1 | 1 | 1 | Trace | 1 | Trace | 1 | Trace | 1 | 1 | 1 | 1 |
| Oxidation Induction Period, hr.* | 7.9 | 9.7 | 7.9 | 10.0 | 10.5 | 9.4 | 10.5 | 15.3 | 7.9 | 14.6 | 7.9 | 7.3 | 10.5 | 6.9 | 10.5 | 7.8 |

*20 ml. of sample at 212 deg. fahr. under oxygen pressure of 100 lb. per sq. in.

air through a weighed drying tube. The weight of water was then obtained directly. The results on the same fuels as used in the previous tests were as follows:

| | Ml. of Water Separated at —20 Deg. Fahr. per 100 Ml. of Fuel Saturated at 77 Deg. Fahr. |
|---|---|
| 100 Per Cent Regular Aviation Gasoline | 0.0005 |
| Aviation Gasoline with 40 Per Cent Benzol | 0.0030 |
| Aviation Gasoline with 40 Per Cent Isopropyl Ether (2 Tests) | 0.0050 and 0.0060 |

The preceding results show that the blend of isopropyl ether deposited hardly twice as much water as the benzol blend and ten times as much as the regular aviation gasoline. Furthermore, in two more tests at —10 deg. fahr. on the blend, in one of which the fuel contained an additional 0.5 per cent of methanol, it was found that, although the same volume of separation occurred in both cases (0.005 per cent), the separation occurred as ice in the absence of methanol and as liquid in the presence of methanol. The melting point of the ice was +25 deg. fahr. and the freezing point of the liquid, which was found to contain about 40 per cent methanol, was —50 deg. fahr. It appears, therefore, that the tendency for isopropyl ether to take up water, although meriting attention under some conditions, is not an important handicap to its use in aviation fuels.

Another characteristic of major practical importance in determining the acceptability of a fuel is its storage stability.

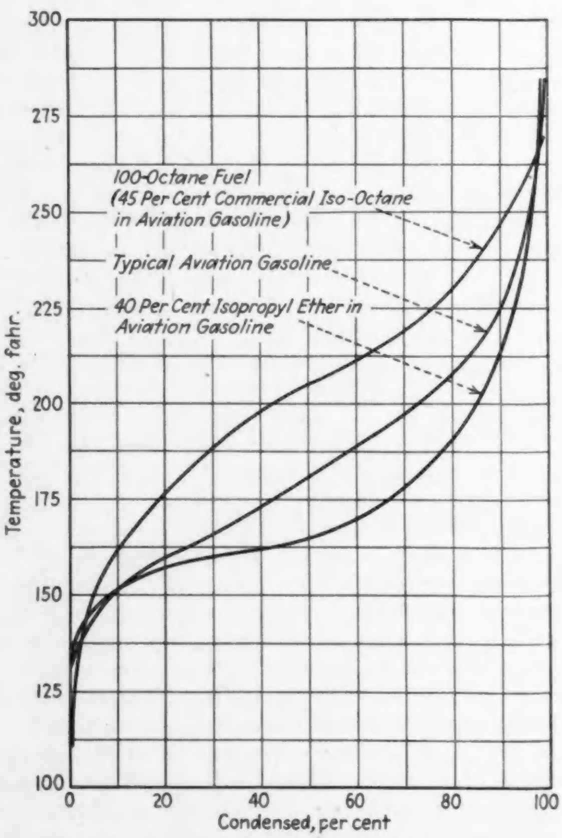



**Fig. 3 – A.S.T.M. Distillation of Aviation Gasolines**

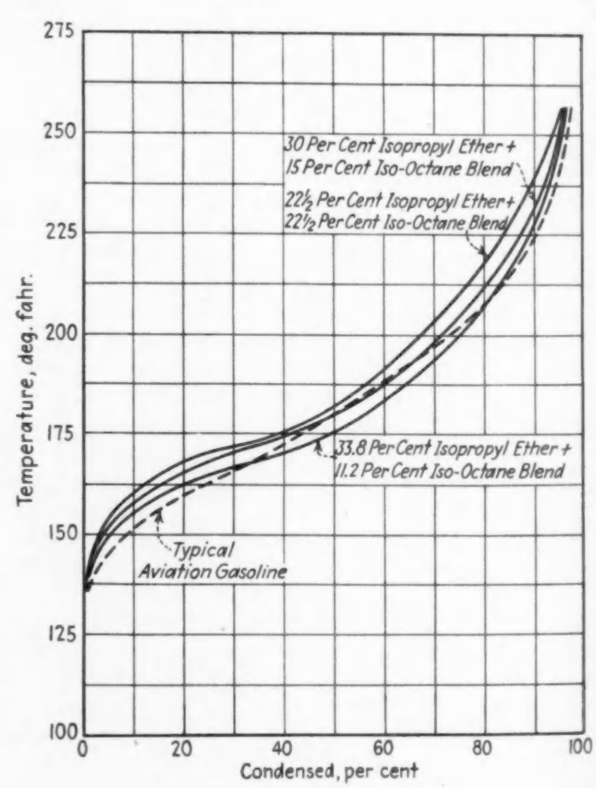



**Fig. 4 – A.S.T.M. Distillation of 45 Per Cent Blends of Mixed Blending Agents in Typical Aviation Gasoline**

Blends of isopropyl ether in aviation gasoline have low gum content by all the usual methods of test. With the addition of 1 to 2 ml. of gasoline inhibitor per 100 ml. of fuel, these blends also have satisfactory resistance to oxidation. The induction period determined at 212 deg. fahr. under oxygen pressure of 100 lb. per sq. in., exceeds 7 hr. The best motor gasolines rarely have an induction period exceeding 6 hr.

The gum tests and the oxidation test usually are accepted as measures of the storage stability of a fuel. However, actual storage tests also are being made to confirm the indications of the accelerated tests. Table 2 shows the preliminary results of storage tests now in progress. After 2½ months at 100 deg. fahr., the stability of the isopropyl-ether blends, both leaded and unleaded, appears unimpaired.

Table 3 gives pertinent inspection data on blended fuels containing 3 cc. of tetraethyl lead and meeting 100-octane number specifications. Specifications for the U. S. Army, U. S. Navy, British Air Ministry, Pratt & Whitney, and Wright Aeronautical Corp., are shown to allow comparison with fuel inspection. A typical aviation gasoline is also included for comparison. It will be observed from Table 3 that a 40 per cent blend of isopropyl ether in aviation gasoline, with 3 cc. of lead per gal. of blend, meets the specifications set by the U. S. Army, U. S. Navy, British Air Ministry, Pratt & Whitney, and the Wright Aeronautical Corp.

Fig. 3 gives the A.S.T.M. distillation for these same blends and the aviation gasoline.

Fig. 4 shows how various 45 per cent combination blends of technical iso-octane and isopropyl ether in aviation gasoline can be used to give a distillation very closely approximating the straight aviation-gasoline distillation.

Fig. 5 shows the distillation effect of various percentages of isopropyl ether in aviation gasoline and demonstrates that mid-range volatility may be varied widely without materially affecting overall boiling range of the gasoline.

## Octane Numbers

A considerable amount of data is available in regard to the octane number and lead susceptibility of blends of isopropyl ether by various methods. Summarized data are shown in Tables 4 and 5, and graphically in Figs. 6 to 9. A study of these results justifies the following conclusions in regard to performance on single-cylinder test engines:

(a) Blended in commercial aviation gasolines, a fuel of 100-

**Table 3 – Characteristics of 100-Octane Aircraft-Engine Fuels**

| | Requirements for Grade 100* | U. S. Navy Requirements | British Air Ministry Requirements | Pratt & Whitney | Wright Aeronautical | New 100-Octane Fuel (40 Per Cent Isopropyl Ether in Aviation Gasoline) | 100-Octane Fuel (45 Per Cent Iso-Octane in Aviation Gasoline) | Typical Aviation Gasoline | Typical Inspection Commercial 100-Octane Aviation Gasoline |
|---|---|---|---|---|---|---|---|---|---|
| Octane Number, Army method | 100 | .. | .. | .. | .. | 100 | 100 | 74 (A.S.T.M.) | 100 |
| Tetraethyl Lead per U. S. gal., ml. | 3 maximum | 3.27 maximum | 3.33 maximum | .. | .. | 3 | 3 | 0 | 3 |
| Copper-Dish Corrosion Test | Must pass | Must pass | .. | Must pass | Must pass | Passes | Passes | Passes | Passes |
| Gum after Accelerated-Aging Test, mg. per 100 ml. | 10 maximum | 10 maximum | 20 maximum | 10 maximum | 10 maximum | 9.8 | 5.0 | 2.0 | 2.0 |
| Sulphur, Per Cent | 0.10 maximum | 0.10 maximum | 0.15 maximum | 0.10 maximum | 0.10 maximum | 0.02 | 0.02 | 0.026 | 0.02 |
| Distillation Test: | | | | | | | | | |
| Per Cent condensed at 150 deg. fahr. | .. | .. | .. | 5 minimum | .. | 8.5 | 5.0 | 9.0 | 9.0 |
| Per Cent condensed at 158 deg. fahr. | .. | .. | .. | 10 minimum | .. | 22.0 | 8.0 | 19.0 | 15.0 |
| Per Cent evaporated at 167 deg. fahr. | 10 minimum | .. | 10 minimum | .. | 10 minimum | 47.5 | 14.0 | 32.0 | 21.5 |
| Per Cent condensed at 167 deg. fahr. | .. | 10 minimum | .. | .. | .. | .. | .. | .. | .. |
| Per Cent evaporated at 212 deg. fahr. | 50 minimum | .. | 50 minimum | .. | 50 minimum | 90.0 | 64.0 | 83.5 | 68.0 |
| Per Cent condensed at 212 deg. fahr. | .. | 50 minimum | .. | .. | .. | .. | .. | 82.5 | .. |
| Per Cent evaporated at 275 deg. fahr. | 90 minimum | .. | .. | .. | 90 minimum | 99.0 | .. | .. | .. |
| Per Cent condensed at 275 deg. fahr. | .. | 90 minimum | .. | .. | .. | .. | .. | 98.0 | .. |
| Per Cent evaporated at 302 deg. fahr. | .. | .. | 90 minimum | .. | .. | .. | .. | .. | .. |
| Sum of 10 and 50 Per Cent Evaporation Points, deg. fahr. | 307 minimum | .. | .. | .. | .. | 319 | 364 | 333 | 354 |
| Per Cent Residue | 2 maximum | 2 maximum | 2 maximum | 2 maximum | 2 maximum | 1.0 | 1.0 | 1.0 | 1.0 |
| Reid Vapor Pressure, lb. per sq. in. | 7 maximum | 7 maximum | 7 maximum | 7 maximum | 7 maximum | 6.0 | 6.0 | 6.6 | 6.7 |
| Freezing Point, deg. fahr. | −76 maximum | −76 maximum | −76 maximum | −76 maximum | −76 maximum | below −148 | below −148 | below −76 | below −76 |
| Water Tolerance: (Ml. change in volume of 20 ml. aqueous layer after shaking with 80 ml. of gasoline at 75 deg. fahr.) | 2 maximum | .. | .. | .. | .. | 0 | 0 | 0 | .. |
| Product of High Value of Heat of Combustion (B.t.u. per lb.) by Specific Gravity | 13,700 | .. | .. | .. | .. | 14,310 | 14,350 | 14,400 | .. |
| Gravity, deg. A.P.I. | .. | .. | .. | .. | .. | 64 | 70 | 70.0 | 70.8 |
| High Heat of Combustion, B.t.u. per lb. | | | | | | 19,360 | 20,820 | 21,000 | |
| Low Heat of Combustion, B.t.u. per lb. | | | | | | 17,990 | 19,410 | 19,580 | |
| Low Heat of Combustion plus Latent Heat of Vaporization: | | | | | | | | | |
| B.t.u. per lb. | | | | | | 18,120 | 19,550 | 19,720 | |
| B.t.u. per gal. | | | | | | 109,400 | 114,500 | 115,100 | |
| Percentage loss in Calorific Value as compared with Typical Aviation Gasoline | | | | | | | | | |
| On lb. basis | | | | | | 8.1 | 0.9 | 0 | |
| On gal. basis | | | | | | 5.0 | 0.5 | 0 | |

*Compare U. S. Army Specification No. 2-92 for Aircraft-Engine Fuel, Grade 100.

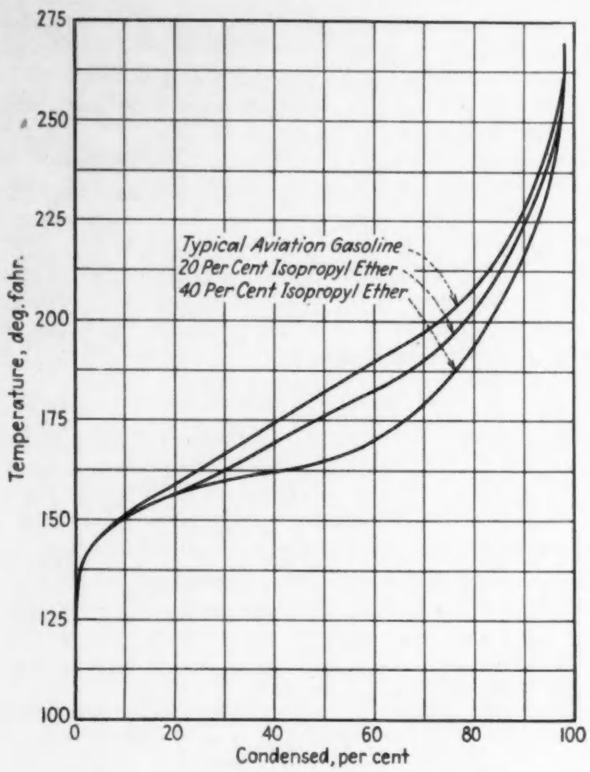



Fig. 5 – A.S.T.M. Distillation of Isopropyl-Ether Blends

octane number (Army method) is possible with 40 per cent of isopropyl ether without exceeding 3 cc. of lead tetraethyl.

(b) The blending value of isopropyl ether is slightly superior to the blending value of technical iso-octane in unleaded blends, and also superior for any fixed amount of lead tetraethyl in the corresponding blends.

(c) The amount of lead tetraethyl necessary to meet a given octane number is less for isopropyl ether than for technical iso-octane in blends containing the same percentage concentrations.

Chemical formulas, boiling points, and octane numbers of various compounds are given in Table 6.

## Economy

The question of the cruising range possible with various fuels naturally raises the question of the limitation of isopropyl-ether blends due to its 5—6 per cent lower heat of combustion on a volume basis. Offhand it would appear that the range would be reduced correspondingly for a given tank capacity. This is not necessarily the case. It has already been pointed out that increasing compression ratio permits of operating at leaner air-fuel mixtures. It has also been shown that the maximum Army octane number of a blend containing not over 3 cc. of lead per gal. is about four points higher for the isopropyl ether than for the corresponding technical iso-octane blend.

Data are lacking on actual performance but, from the best figures available, it would appear that compression ratio could be increased—or possibly supercharger boost increased—to permit of operating at about a 10 per cent lower specific fuel consumption, thus compensating for the lower heating value. There are indications also that, for equal 100-octane (Army rating) fuels, the isopropyl-ether blend will permit of a leaner mixture without exceeding the critical head temperature and, providing compression ratio is high enough to permit of very lean mixtures, again compensating for its in-

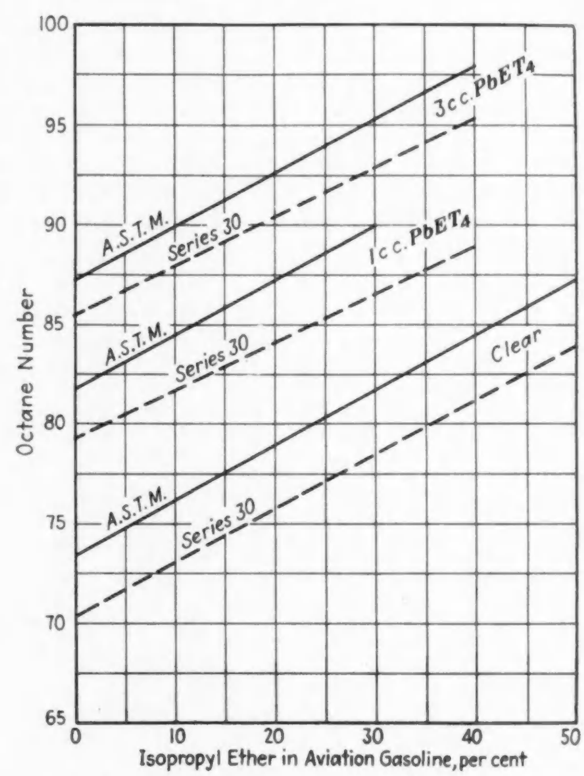



Fig. 6 – A.S.T.M. and Series 30 at 375 Deg. Fahr. Ratings of Clear and Leaded Blends of Isopropyl Ether in Aviation Gasoline

Table 4 – Single-Cylinder Octane-Number Data for Blends in Aviation Gasoline of 74 C.F.R.-M.M.

| Blending Agent | Volume Per Cent Added | A.S.T.M. Octane Numbers: Clear | 1 Cc. Ethyl Lead per Gal. | 3 Cc. Ethyl Lead per Gal. | Clear Blending Value | Army Octane Number 3 Cc. Ethyl Lead per Gal. | Series 30 at 375 Deg. Fahr. Octane Numbers Clear |
|---|---|---|---|---|---|---|---|
| Isopropyl Ether | 0 | 73.5 | 82.0 | 87.5 | .... | .... | 69 |
| | 10 | 76.3 | 84.1 | 89.3 | 102 | .... | 72 |
| | 25 | 80.5 | 88.6 | 94.0 | 101 | .... | 78 |
| | 40 | 85.0 | .... | 97.5 | 102 | 100 | 82 |
| Iso-Octane | 0 | 73.5 | 82.0 | 87.5 | .... | .... | 69 |
| | 10 | 76.0 | 84.0 | 89.0 | 99 | .... | 72 |
| | 25 | 79.4 | 86.8 | 91.4 | 97 | .... | 76 |
| | 40 | 83.0 | .... | 94.2 | 97 | 99 | 80 |
| | 50 | 85.5 | 93.0 | 96.0 | 98 | .... | 82 |

**Table 5 – Single-Cylinder Army Octane Numbers of Blends of Isopropyl Ether and Technical Iso-octane in Knock Reference Fuel C-9**

| Blends of Reference Fuel C-9: Volume Per Cent of Isopropyl Ether | Volume Per Cent of Crude Iso-octane | Octane Numbers by Army Method: Clear | With 3 Cc. Tetraethyl Lead per Gal. | Clear Blending Value of Blending Agent |
|---|---|---|---|---|
| 0 | 0 | 74.8 | 93.0 | .... |
| 10 | 0 | 79.0 | 94.0 | 117 |
| 20 | 0 | 82.7 | 96.6 | 114 |
| 30 | 0 | 85.7 | 98.4 | 111 |
| 40 | 0 | 88.3 | Equals iso-octane + 0.18 cc. lead | 109 |
| 50 | 0 | 90.6 | Ditto + 0.44 cc. lead | 106 |
| 60 | 0 | 93.5 | Ditto + 0.82 cc. lead | 106 |
| 5 | 5 | 77.8 | 93.6 | 105 |
| 10 | 10 | 81.5 | 96.0 | 108 |
| 15 | 15 | 84.1 | 97.6 | 106 |
| 20 | 20 | 86.7 | 99.2 | 105 |
| 25 | 25 | 88.8 | Equals iso-octane + 0.08 cc. lead | 103 |
| 30 | 30 | 91.0 | Ditto + 0.22 cc. lead | 102 |
| 0 | 30 | 82.0 | 97.0 | 99 |
| 0 | 40 | 84.4 | 98.7 | 99 |
| 0 | 50 | 86.5 | Equals iso-octane + 0.05 cc. lead | 98 |

(Data obtained through the courtesy of S. D. Heron, Ethyl Gasoline Corp.)

herently lower heating value. Needless to state, this discussion applies only to comparisons with 100 per cent hydrocarbon fuels of 100-octane number (Army rating). The isopropyl-ether blends of 100-octane fuels (Army rating) are far superior to all fuels of lower octane number.

## Availability

Of particular interest when proposing a new and different fuel is the question of potential supplies of raw materials as related to potential demand. The total U. S. consumption for all grades of aviation fuels was 63,000,000 gal. for 1934 and is estimated as 88,500,000 gal. for 1936.

A recent survey of potential supplies of propylene, the raw material for making isopropyl ether, has shown that in this

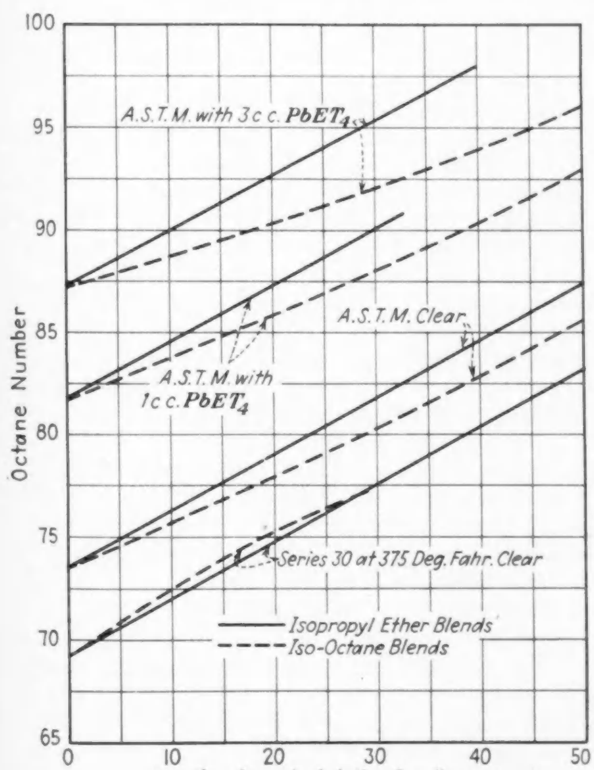



Fig. 7 – A.S.T.M. and Series 30 at 375 Deg. Fahr. Ratings of Clear and Leaded Blends of Isopropyl-Ether and Iso-Octane Blends in Aviation Gasoline

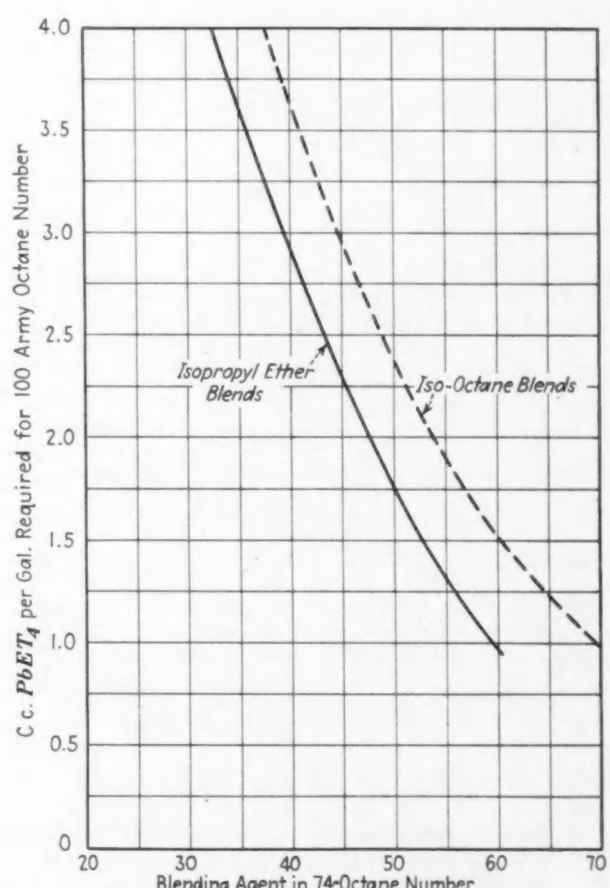
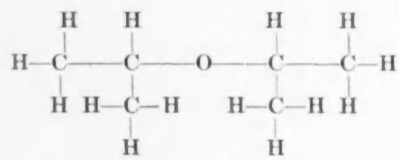
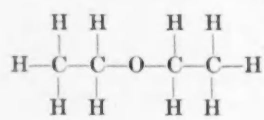
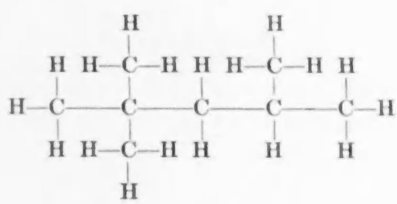
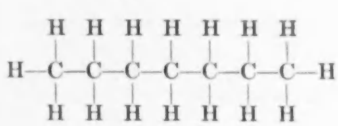
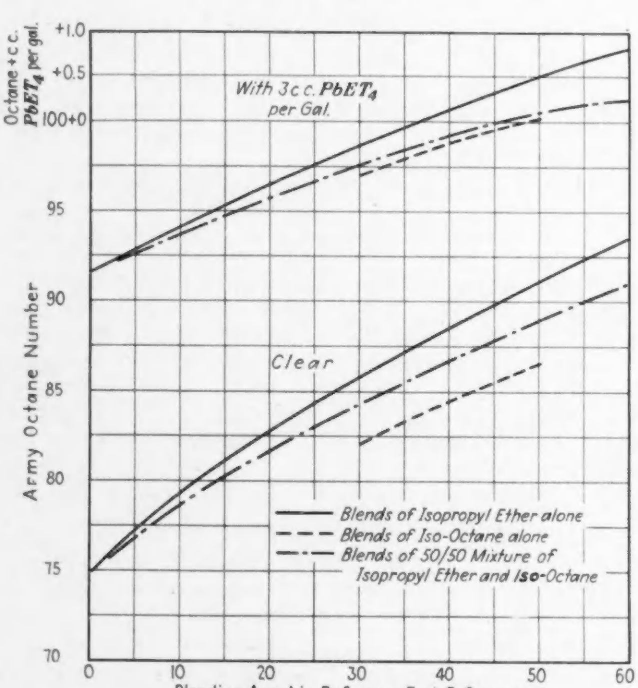



Fig. 8 – Lead Requirements of Aviation-Gasoline Blends for 100 Army Octane Number

**Table 6 – Chemical Formulas, Boiling Points, and Octane Numbers of Various Compounds**

| Compound | Chemical Formula | Boiling Point, deg. fahr. | A.S.T.M. Octane Number (Clear Blending Value When Blended 25 Per Cent in Aviation Gasoline) |
|---|---|---|---|
| Isopropyl Ether ($C_6H_{14}O$) |  | 154 | About 100 |
| Ethyl Ether ($C_4H_{10}O$) |  | 94 | Below 0 |
| Iso-Octane ($C_8H_{18}$) |  | 211 | 100 |
| n Heptane ($C_7H_{16}$) |  | 211 | 0 |

country, sufficient of it is available, exclusive of all other normal demands for other purposes, to produce at the present time approximately 340,000,000 gal. per year of technical isopropyl ether. Assuming that 100-octane fuel is required (using 3 cc. of lead per gal.), this amount is sufficient to produce some 850,000,000 gal. per year of finished product. Thus it is apparent that the availability of this fuel is much more than sufficient to meet the entire requirements for 100-octane gasoline even in the event of National emergency. When combined with the available supplies of technical iso-octane, estimated at 155,000,000 gal. per year, there is every assurance that adequate quantities will be immediately available for any demand. The cost of the material for large production is entirely reasonable.




*Courtesy of S. D. Heron, Ethyl Gasoline Corp.*

**Fig. 9 – Octane Number Blending Characteristics of Isopropyl Ether and Iso-Octane by Army Method of Test**

When isopropyl ether and iso-octane are available, blends of the two offer certain advantages lacking in either when used separately, particularly with respect to volatility. Data have been presented which show that a 50-50 blend of the two in standard knock reference fuel C-9 has an antiknock value superior to that obtained when using iso-octane alone as the blending agent. By using such blends a fairly wide range of volatilities is made available to meet special requirements. In cases where higher front-end volatility is desired, a material such as iso-pentane can be used without any material sacrifice in antiknock value.

### Multicylinder Tests

Tests have been run on high-output multicylinder engines comparing the performance of the following fuels:

- Straight-run gasoline + iso-octane + lead to 100-octane number.
- Straight-run gasoline + isopropyl ether + lead to 100-octane number.
- Straight-run gasoline + lead to 92-octane number.

*(Continued on page 357)*

# Air-Cooled Radial Aircraft-Engine Installation

**By P. A. Anderson**
*Field Engineer, Wright Aeronautical Corp.*

**THIS paper deals with air-cooled aircraft-engine installations and covers such units as cowling, engine mounts, exhaust systems, carburetor-ice eliminators, oil systems, fuel systems, controls, accessories, and so on.**

**No attempt is made to describe any particularly new ideas in engine-installation design, but rather the paper explains the fundamental requirements for satisfactory operation of Wright air-cooled aircraft engines.**

DURING the past decade, and notably during the development of commercial air transport in this country, improvement in airplane performance has exceeded expectations. This improvement has been the result of greater power and increased efficiency of practically all elements of the airplane. The high resistance of the radial engine was greatly reduced by the development of the N.A.C.A. cowl. In the case of multimotored airplanes, the efficiency was further increased by better nacelle locations as determined by extensive experimental study conducted by the National Advisory Committee for Aeronautics. During this period of development, careful attention to engine-installation requirements has contributed in no small degree to the reliability of the modern powerplant, and it is on this subject that this paper is presented.

### A Typical Installation

Because of low weight per horsepower when completely installed, together with its reliability and economy, the air-cooled radial engine has been chosen to power the greater part of the modern aircraft in this country. As the power output has increased, the non-uniformity of the operating characteristics of each engine model has become more pronounced and installation requirements have, of necessity, had to be more specific.

To avoid the complications of these variations the Wright Cyclone Model GR-1820G-2 engine has been selected for discussion (see Fig. 1). The dry weight of this engine is 1163 lb., which weight includes 16:11 reduction gearing, magnetos, radio-shielded spark-plugs and ignition, automatic mixture control, carburetor, automatic valve-gear lubrication, propeller-hub attachment parts, cylinder-head and barrel baffles, exhaust-stack flanges and drives for the generator, starter, fuel pump, vacuum pump, constant-speed propeller governor, two gun synchronizers, landing-gear hydraulic pump, robot pilot pump, and tachometer. An installation-weight analysis of a typical modern air transport equipped with this engine is given in the following table:

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 4, 1936.]

Typical Installation Weights of GR-1820G-2 Engine
(all parts forward of firewall)

| Item | Weight, Lb. |
|---|---|
| Engine | 1163.0 |
| Propeller | 371.0 |
| Cowling (inner and outer) | 71.0 |
| Engine Mount | 51.0 |
| Exhaust Collector and Carburetor-Air Heater | 87.5 |
| Carburetor-Air Inlet Duct and Control Valves | 22.5 |
| Oil Cooler, Supports, and Air Duct | 30.0 |
| Oil Lines and Fittings | 13.0 |
| Fuel Lines and Fittings | 8.5 |
| Generator | 35.0 |
| Starter | 32.0 |
| Miscellaneous Other Items | 27.5 |
| Total | 1912.0 |

The engine has a take-off rating of 1000 hp. at 2200 r.p.m. and a critical-altitude rating of 850 hp. at 2100 r.p.m. at 5800 ft. (as shown in Fig. 2). This power is obtained when operating with standard conditions, namely, 60 deg. fahr. carburetor air; 29.92 in. hg. barometer at the carburetor-air inlet and cylinder-exhaust outlet; 75 per cent vapor pressure; best-power mixture settings; not over 400 deg. fahr. maximum rear spark-plug-washer temperature; and 300 deg. fahr cylinder-base temperature. Variations from these standard conditions will affect the power output as shown in Fig. 3. Naturally, changes in barometer and vapor pressure are beyond control, but all other operating conditions can be controlled largely by satisfactory engine installation.

### Engine Cooling

Probably the most critical installation problem is that of engine cooling. Ironically, two of the most distinguished contributions to airplane performance have complicated the cooling problem. First, the reduction of drag resulting from the smaller N.A.C.A. cowl-front opening and, second, the

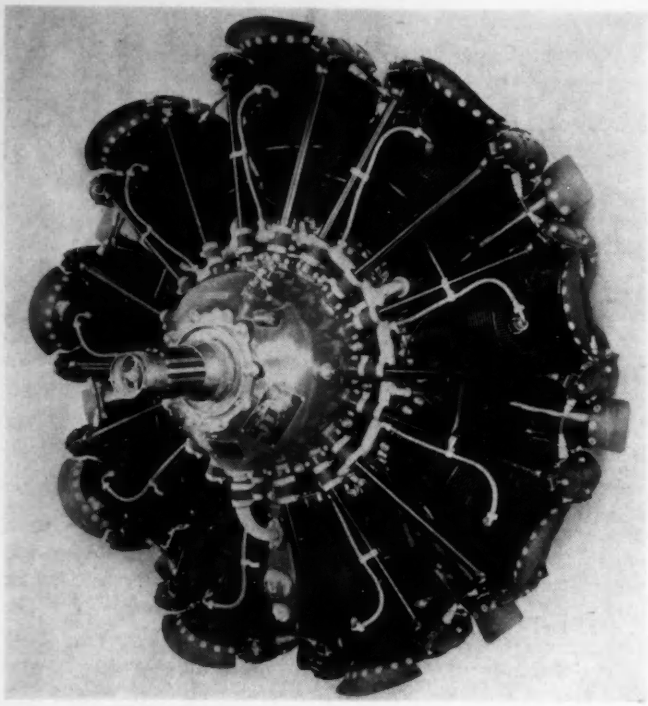


Fig. 1 – Wright Cyclone Engine, Model GR-1820G-2

less-effective propeller blade of the controllable-pitch propeller within the cowl opening, have greatly reduced cooling-air flow.

Fig. 4 shows this trend over a period of years. In 1928 the average Cyclone engine installation cowl-front opening was equal to 85 per cent of the frontal area of the engine. In the average new 1936 Cyclone engine installation this figure has dropped to 55 per cent. In 1928 the percentage of frontal area of the engine which was washed by the effective flat blade of the propeller was 60 per cent whereas, in 1936 due to reduction in cowl opening and movement of the effective flat blade away from the propeller hub, this figure has dropped to 25 per cent. Throughout this period of drag reduction and increase in propeller efficiency the engine horsepower has increased more than 80 per cent without material gain in the flying speed at which rated power is used. (This period of engine development is described in the paper[1] presented at the Annual Meeting of the Society, Detroit, January, 1936, by Raymond W. Young.)

Fig. 5 illustrates the average N.A.C.A. cowl form now in use and indicates by arrows the characteristic air forces that act upon the cowl surface. This design has now reached a stage where air-flow within the cowl is inadequate at certain air speeds unless auxiliary devices are used to produce the air-flow that formerly was supplied by the propeller. Most notable of these devices are the nose-cowl baffle and the adjustable-ring-cowl trailing edge, both of which have been objectionable from the standpoint of airplane maintenance and performance.

In an effort to improve cooling the Wright Aeronautical Corp. designed and are constructing for flight test, a reverse-flow cowl, the general lines of which are shown in Fig. 6. It will be noted that cooling air is discharged at the low-pressure area of the ring-cowl leading edge, which arrangement is expected to accelerate cooling-air flow. The effect on airplane drag is problematical but, by eliminating the drag and air-flow disturbance caused by the N.A.C.A. ring-cowl trailing-edge slot, no serious reduction of airplane speed is anticipated. Results of tests are not yet available.

Regardless of the design it must be remembered that the cowl is responsible for the satisfactory cooling of many parts of the engine beside cylinders. Items such as spark-plug elbows, magnetos, fuel lines, exhaust collectors, intake pipes, carburetor-air and oil-cooler-air inlet ducts, must be protected against excessive heat, the temperature limitations of which are listed later under the heading, "Ground and Flight Tests." These temperature limitations may be controlled by supplying the minimum cooling air-flow requirement as determined for all engine models, the evaluation of which for this engine is 1 in. $H_2O$ baffle-pressure differential for ground running at 1300 r.p.m. on propeller load and 3 in. $H_2O$ baffle-pressure differential for rated power and speed. Fig. 7 shows a method of mounting the cowl in rubber, utilizing the mounting lugs on the engine rocker boxes.

## Engine Mount

A unit that must receive most careful consideration is the engine mount. It is generally conceded that, from the standpoint of weight, strength, reliability, accessibility, and cost, the tubular-steel mount is the most satisfactory type. Aside from the required strength factors, the mount should be designed to provide a rigid structure with members arranged so that all engine accessories will be available for removal, inspection, and adjustment. It should be securely attached and easily removable at the fire wall. Most important, it should have adequate provision for vibration absorption. Fig. 8 shows a mount that has proved satisfactory in service. Rubber bushings installed at each engine-mounting lug allow adjustment of the amount of flexibility by varying the rubber density and section should the natural vibration frequency of the complete structure fall within the speed range of engine operation.

The primary and secondary engine unbalance and the half order torsional variation due to gas torque reaction of the GR-1820G-2 engine is not objectionable in an airplane whose structure does not have the frequencies of its modes of free vibration coincident with the frequencies of the exciting forces from the engine or propeller. Considerable information may be obtained on the resonant response of the various modes of free vibration of the aircraft structure by exciting vibration with an unbalanced rotating weight on the nose of the engine. Fig. 9 shows the vibrator assembled on nose of airplane. Fig. 10 shows the details of this equipment consisting of a special propeller-hub nut which supports a ball-bearing-mounted unbalance weight which, in turn, is driven through a rubber hose by a variable-speed motor having a speed range of 600 to 5000 r.p.m. The test procedure is as follows:

Adjust the unbalanced weight to produce the amplitude desired when operated through the entire speed range from 600 r.p.m. to twice the over-speed rating of the engine. With each increment of speed take amplitude records of the various seats in the airplane and any other parts of the structure which have objectionable vibration. The indicator shown in Fig. 10 is excellent for this purpose. It is a standard dial indicator mounted on a steel block with sponge-rubber base.

Before flight test the propeller should be checked very accurately as to its balance, track, and pitch of individual blades, and the engine should be checked carefully as to proper carburetion and ignition. There should be no rigid

[1] See S.A.E. Transactions, June, 1936, pp. 234-256; "Air-Cooled Radial Aircraft-Engine Performance Possibilities," by Raymond W. Young.

attachments between the engine and fuselage, such as exhaust-collector supports, cowling, controls, and plumbing. A substitute for the variable-speed motor used during the ground test is a reed tachometer shown in Fig. 10. The tachometer should have sufficient speed range to indicate propeller-blade pulsation or second-order engine vibration, whichever is the greater. In flight operate the engine through the normal operating range, noting with the aid of the engine tachometer the engine speed at which frequencies are indicated on the reed tachometer together with the amplitude as measured with the indicator. Usually only those parts which the pilot or passengers contact such as seats, controls, and so on need be measured.

Fig. 11 shows a curve that summarizes the results of subjecting a great many individuals to a vibration test rig and of obtaining their impression of the threshold of unpleasant vibration. This curve is reported in Memoranda No. 1637 of the Aeronautical Research Committee reports by Mr. Constant. When the amplitude of the observed vibration exceeds the curve shown on Fig. 11, the vibration may be considered objectionable provided

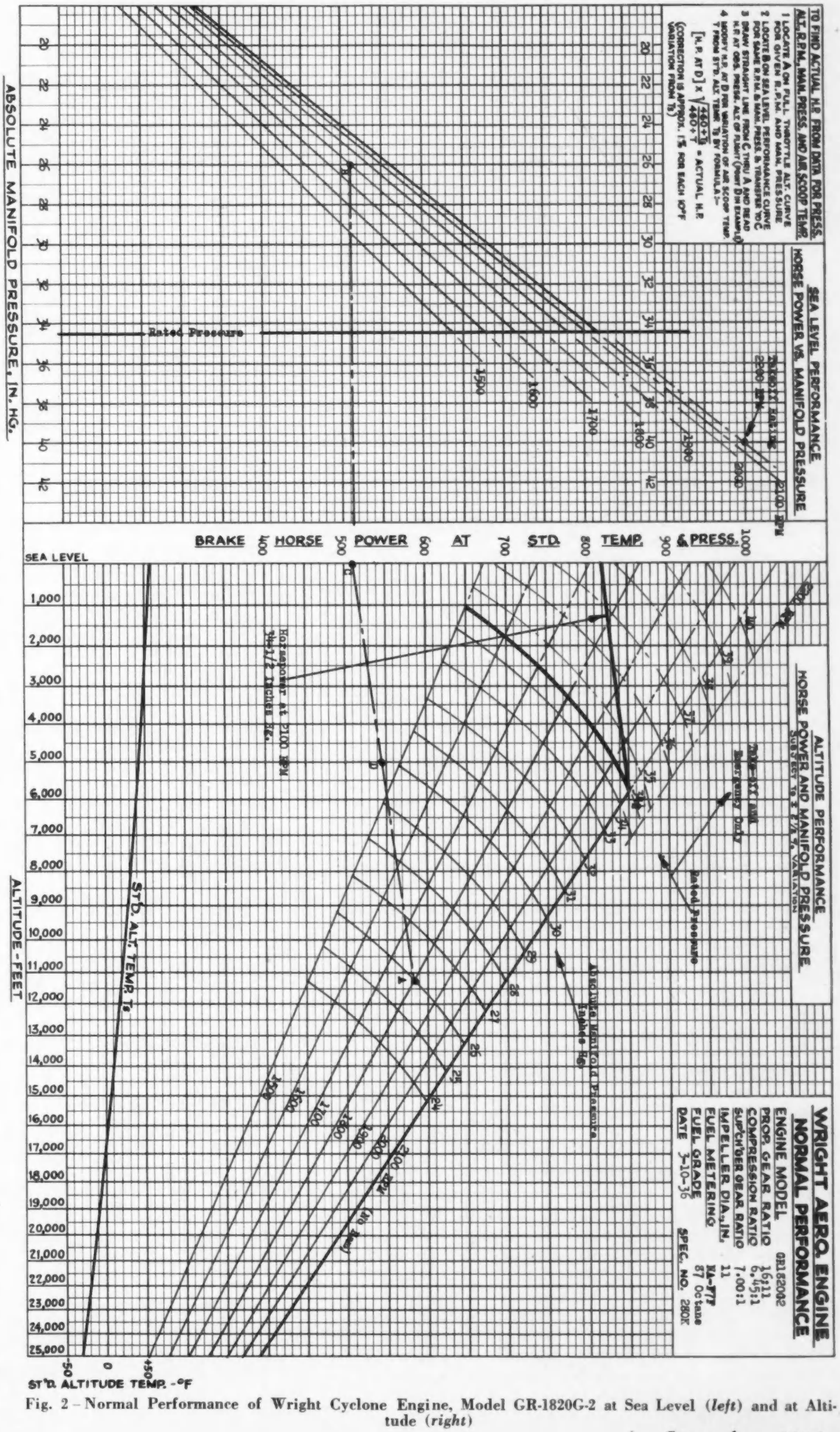



Fig. 2 – Normal Performance of Wright Cyclone Engine, Model GR-1820G-2 at Sea Level (*left*) and at Altitude (*right*)

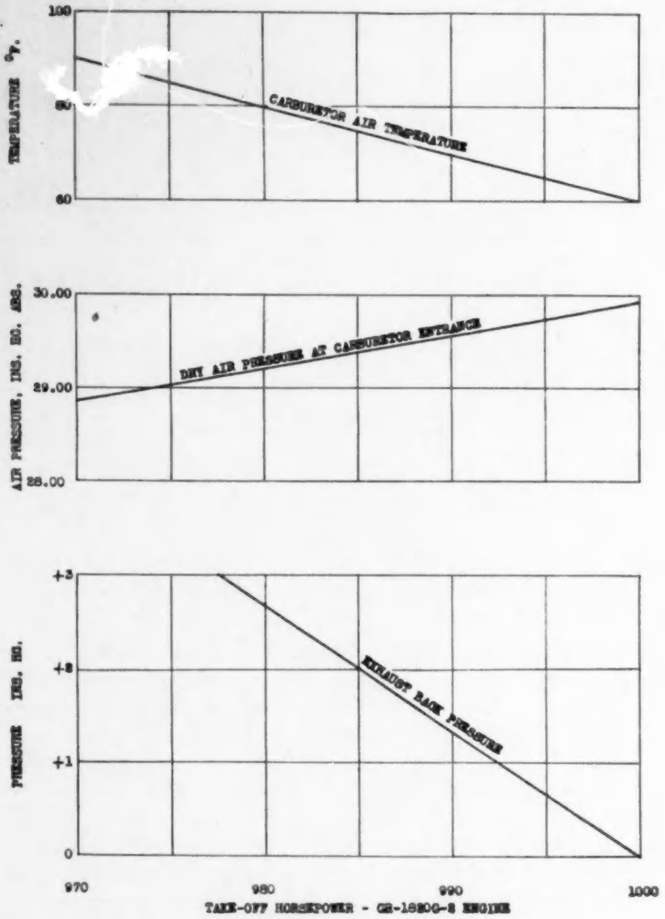



**Fig. 3 – Variation in Horsepower with Changes in Operating Conditions**

Standard Conditions
Cylinder-head temperatures—400 deg. fahr. maximum.
Cylinder-base temperatures—300 deg. fahr. maximum.
Mixture strength for best power.

the frequencies occur within the operating-speed range.

Change in the elasticity of the engine rubber-mount bushings usually causes a greater alteration in the vibration characteristics than any other one modification. In general, it may be stated that increased elasticity, as provided by rubber, usually reduces both the frequency and amplitude of objectionable vibration. Three-bladed propellers usually produce less vibration disturbance than those of two blades, especially if the objectionable vibration is confined to climbs and turns. Lack of rigidity in the fuselage structure behind the engine mounting, either flexuously or torsionally, is liable to produce unpleasant roughness of low frequency, which can be eliminated only by bracing or stiffening the structure.

## Carburetor-Air Heater

The development of means for preventing ice formation in the carburetor, especially during recent years when improved flying equipment has allowed flight operation through rain, sleet, and snow, is the result of research from many sources. The cause of ice formation is, to a large extent, a basic fault in the carburetor design. Vaporization of the fuel as it leaves the discharge nozzles of the carburetor causes a temperature drop of 40 to 80 deg. fahr. at this point, depending upon the volatility of the fuel, fuel-air mixture ratio, and other factors. When the incoming air is of a certain temperature and contains sufficient moisture, ice forms on the throttle butterfly and loss of power results. The difficulty was overcome by installing carburetor-air preheaters capable of maintaining 100 deg. fahr. air to the carburetor while operating below a 0 deg. fahr. atmospheric temperature.

A typical heater is shown in Fig. 12. The heating element consists of two 3-in. diameter tubes installed within the exhaust collector and connected to the carburetor-air inlet duct. Suitable valves control the temperature of the air entering the carburetor and by-pass the excess heated air to provide ventilation for the tubes when only cold air is required. Preheated carburetor air causes power loss as shown in Fig. 3 and, if excessive, it will result in detonation, high cylinder and oil temperatures, and burned pistons, all of which will contribute to a major engine failure if allowed to exist for long. The fuel-air mixture temperature indicator shown in Fig. 13 provides a means for limiting the amount of preheat to the minimum required. When atmospheric conditions are such that carburetor ice formation occurs, the pilot need only apply sufficient heat to maintain an above-freezing temperature (35 to 40 deg. fahr.).

Considerable progress is being made in the development of a carburetor in which no parts are exposed to the fuel-air mixture and the resultant low temperature. Extensive tests conducted by the Wright Aeronautical Corp. have proved

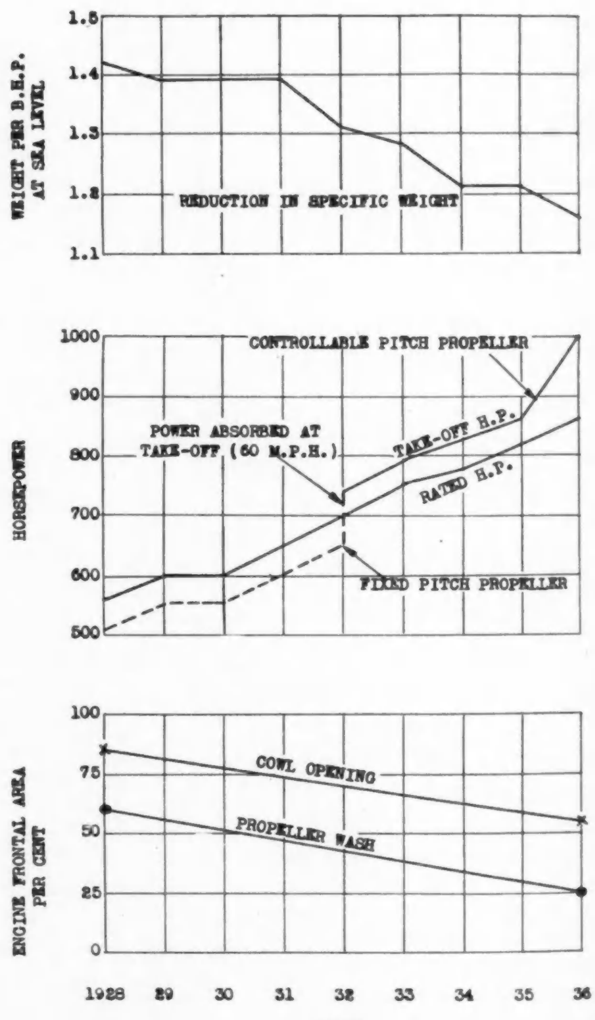



**Fig. 4 – Concurrent Improvements in Cyclone Engine Characteristics and Decrease in Cooling Provided**

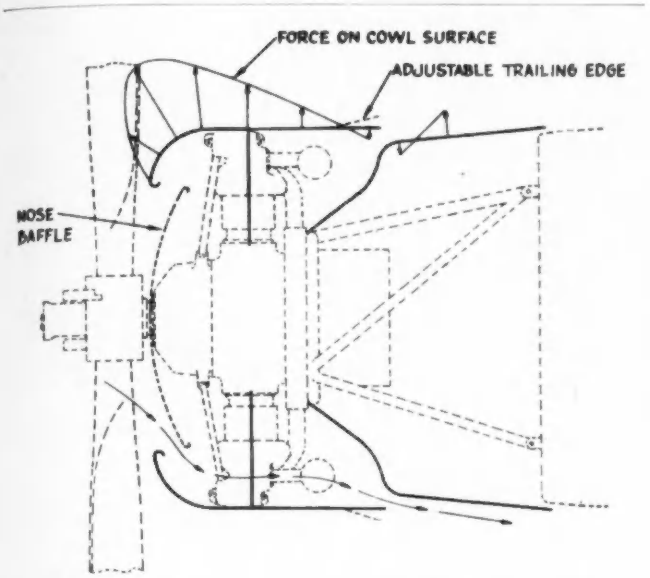



Fig. 5 – Normal Type of N.A.C.A. Cowl

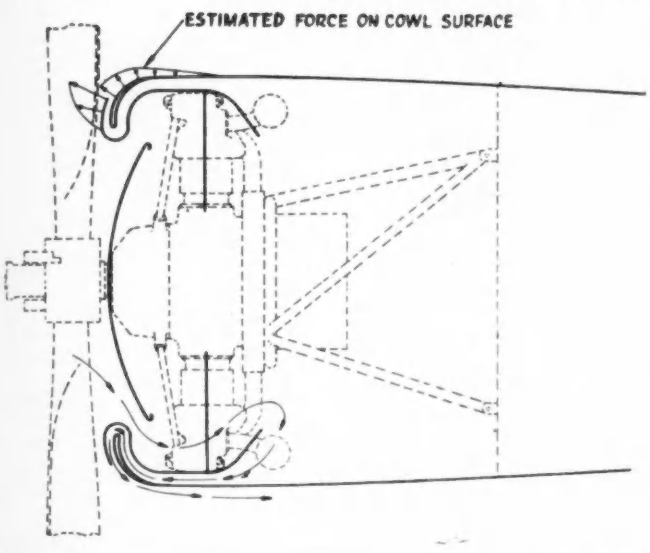



Fig. 6 – Wright Reverse-Flow Cowl

quite conclusively that icing will not occur in this type of carburetor except during the comparatively rare occasions when atmospheric conditions are such that ice forms on airplane wings, at which time a few degrees of preheating will be sufficient to eliminate the carburetor ice. Details of this carburetor are not yet available for publication.

### Exhaust Collector

Exhaust-collector back-pressure causes loss of power (as shown in Fig. 3), faster deterioration of the material, and excessive noise. The resultant high temperature of the entire exhaust collector will contribute to overheating of the vital accessory units in the engine rear compartment. Careful design will result in negative pressure within the entire collector while operating with maximum power. The maximum back-pressure should not exceed 6 in. $H_2O$. The following formula is a satisfactory guide for the net cross-sectional area required: Net area (in.$^2$) $=$ 0.04 x maximum hp. per cylinder x number of cylinders exhausting into the section. When heater pipes are inserted within the collector, the constant should be increased to 0.045 to compensate for added friction.

The exhaust discharge from each cylinder should be as near tangential to the flow in the collector as is possible. The collector should be located well away from the engine crankcase, preferably in line with the exhaust outlets to insure adequate cooling-air flow and no part of it, other than the cylinder connections, should be within 1½ in. of any part of the engine, greater clearance being preferable for the ignition wiring. Because of uniform expansion and contraction, a round section is advisable. Expansion joints should be provided between all cylinders. Welded joints should be no less than ½-in. radius to avoid sharp corners and subsequent failures due to local hot-spots. All welded joints should be smooth within the collector. Fig. 12 shows a typical collector.

### Induction System

One of the most important units in the induction system is the carburetor air-inlet duct, the principal function of which is to provide maximum ram, minimum temperature rise, and non-exposure to dirt "whipped up" from the ground by the propeller. Fig. 14 shows the gain in engine performance that may be obtained by taking full advantage of ram. For instance, the critical altitude of the GR-1820G-2 engine would be increased from 5800 ft. to 7200 ft. in an airplane with a top speed of 250 m.p.h. at 5800 ft. if theoretical ram could be obtained. Generally, only 80 per cent of the theoretical ram is obtained with a well-designed air-inlet duct. Points of maximum ram, minimum temperature, and minimum entrapping of dirt are on the upper side of the N.A.C.A. cowl just forward of the ring-cowl trailing edge and within the cowl forward of the upper cylinders. Both locations are favorable for the installation of the air duct to the down-draft carburetor. The air-duct area should be at least 10 per cent greater than the combined area of the carburetor venturi. Suitable connections and valves should be provided for the carburetor air-heater system.

### Fuel System

The function of any fuel system is to provide an uninterrupted supply of fuel to the carburetor at the proper pressure and temperature under all conditions of engine operation. Figs. 15 and 16 show typical single-engine and twin-engine fuel systems respectively. Lines from each tank are

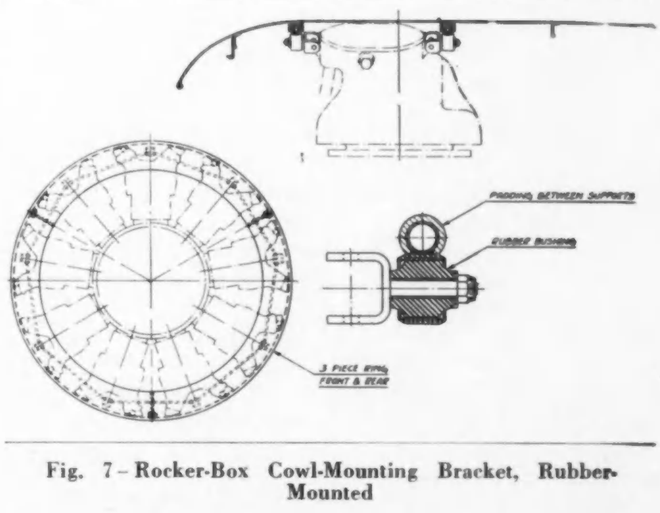


Fig. 7 – Rocker-Box Cowl-Mounting Bracket, Rubber-Mounted

connected to a selector valve. The combined strainer, hand pump, and relief valve is being used extensively as a means for reducing the number of joints, fittings, and tubing and for making available more space in the aircraft. The fuel pump, containing bypass and pressure-relief valves, may be assembled directly to the engine or driven remotely. Remote drives are used to "drown" the pump by placing it near the bottom of the fuel supply, thus reducing the pump suction and fuel temperature which, in turn, reduces the possibility of air and vapor lock.

The Department of Commerce specifies that all fuel piping

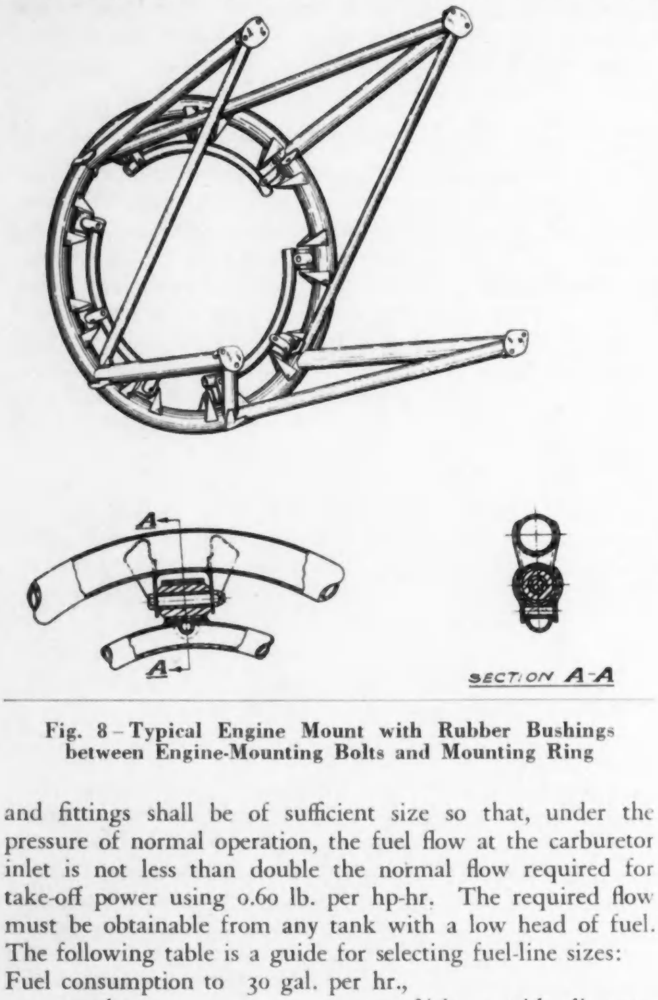



**Fig. 8 – Typical Engine Mount with Rubber Bushings between Engine-Mounting Bolts and Mounting Ring**

and fittings shall be of sufficient size so that, under the pressure of normal operation, the fuel flow at the carburetor inlet is not less than double the normal flow required for take-off power using 0.60 lb. per hp-hr. The required flow must be obtainable from any tank with a low head of fuel. The following table is a guide for selecting fuel-line sizes:

Fuel consumption to 30 gal. per hr.,
or 300 hp. . . . . . . . . . . . . . . . . . . . . . . . . . 3/8-in. outside diameter
Fuel consumption to 60 gal. per hr.,
or 600 hp. . . . . . . . . . . . . . . . . . . . . . . . . . 1/2-in. outside diameter
Fuel consumption to 100 gal. per hr.,
or 1000 hp. . . . . . . . . . . . . . . . . . . . . . . . . 5/8-in. outside diameter

The preceding sizes are for plain tubing and represent the inside diameter required for flexible tubing.

Air lock comes about through the locking of air within the system, whereas vapor lock comes from the formation of fuel vapor within the system. The boiling point of gasoline may be lowered by reducing the atmospheric pressure on it, as happens when the effect of pump suction or inlet depression is imposed on the gasoline. A combination of high suction and high temperature will induce vapor lock sooner. A third factor hastening the formation of fuel vapor lock is altitude which, because of the lower atmospheric pressure, causes fuel to boil at lower temperature. Therefore, the usual

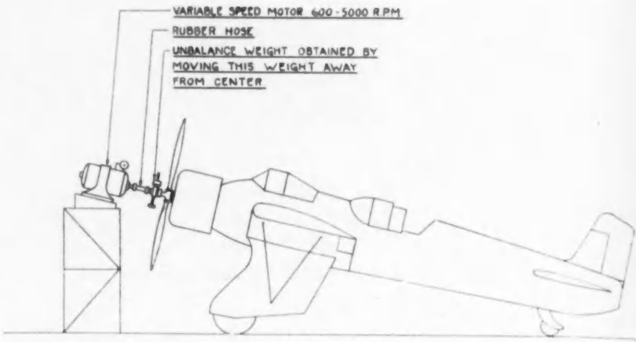



**Fig. 9 – Vibrator Assembly on Nose of Airplane**

aviation gasolines with 10 per cent boiling temperatures around 140 deg. fahr. may cause considerable trouble from vapor locking when no provision is made to keep the fuel temperatures below their critical points. Vertical bends in the fuel lines should be avoided. It is advisable to use a fuel pump that is capable of high suction, such as the vane type, in order to exhaust trapped air from the lines quickly. Fuel pressure should be measured at the connection provided on the carburetor. When the capillary-tube type pressure gage is used, the gage reading should be corrected for the difference in level between the carburetor and the gage by subtracting 0.3 lb. per sq. in. for each foot of head on the gage.

## Oil System

The standard oil system consists of an oil tank connected to the engine by inlet and outlet oil lines and a vent line which returns air to the engine discharged through the scavenging system. Temperature is regulated by an oil cooler located in the discharge line from the engine. Heat rejection to oil in

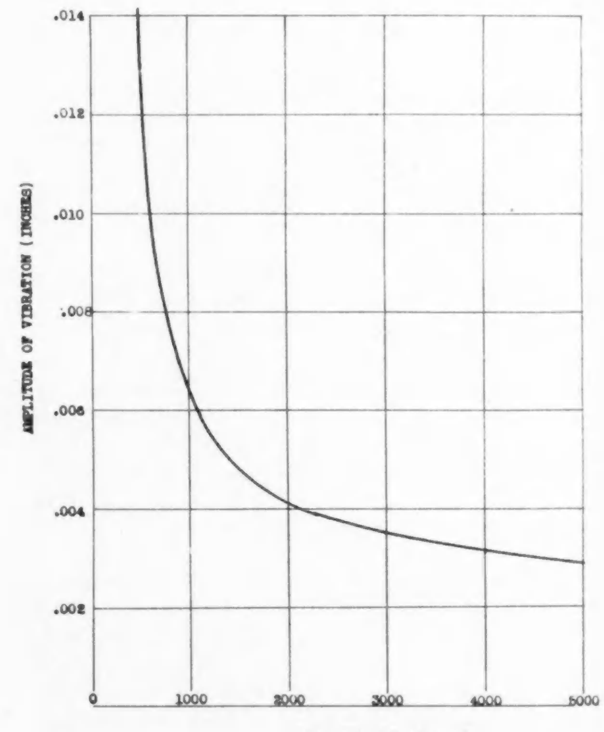



**Fig. 11 – Relation between Frequency and Amplitude Defining the Threshold of Unpleasant Vibration**

From Aeronautical Research Committee Reports and Memoranda No. 1637 by Constant.

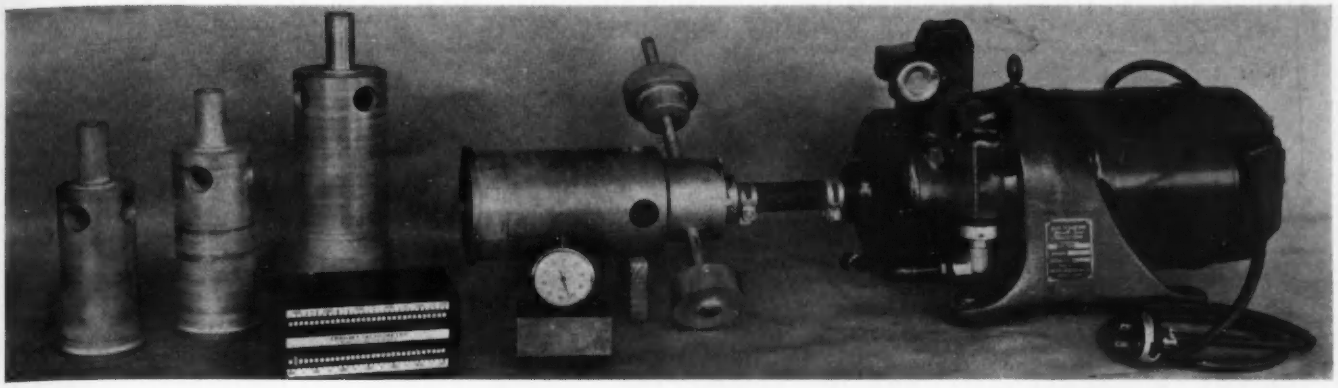


Fig. 10 – Details of Vibrator

the GR-1820G-2 engine is approximately 1400 B.t.u. per min. and, unless this heat is dissipated properly, scavenging troubles and loss of power result.

The Department of Commerce requirement of 1 gal. of oil for every 16 gal. of fuel plus the minimum specified for safe operation of the engine by the engine manufacturer allows sufficient oil supply for normal maximum consumption. For Wright engines the minimum required supply is 2 gal. to 300 hp.; 3 gal. to 500 hp.; 4 gal. to 700 hp., and so on, the amount being proportioned to the heat rejection to the oil. At least 10 per cent expansion space is provided in the oil tank for possible uneven circulation and foaming.

The cooler should have provision for maintaining viscosity low enough to insure flow through the cooler cores when operating in low atmospheric temperatures. An oil-temperature regulator should be provided either independent or as a part of the oil cooler. Normally, this regulator is the sylphon type which by-passes the cooler when oil temperature is below the recommended limits of 140 deg. fahr. to 160 deg. fahr.

The electrical system consisting of wiring, shielding, starter, generator, battery, magneto, booster-magneto, spark-plugs, ignition switch, and thermocouples, requires careful consideration to guard against grounding and open and short circuits. It is desirable to install ignition and other electrical wires on the side of the engine compartment opposite to the fuel lines in order to decrease the fire hazard. All high- and low-tension circuits should be grouped separately and placed within conduits to afford protection as well as shielding. Jack boxes at the fire wall should be provided to permit easy engine removal.

Controls installed to operate the throttle, mixture, spark advance, propeller pitch and speed, and carburetor heater

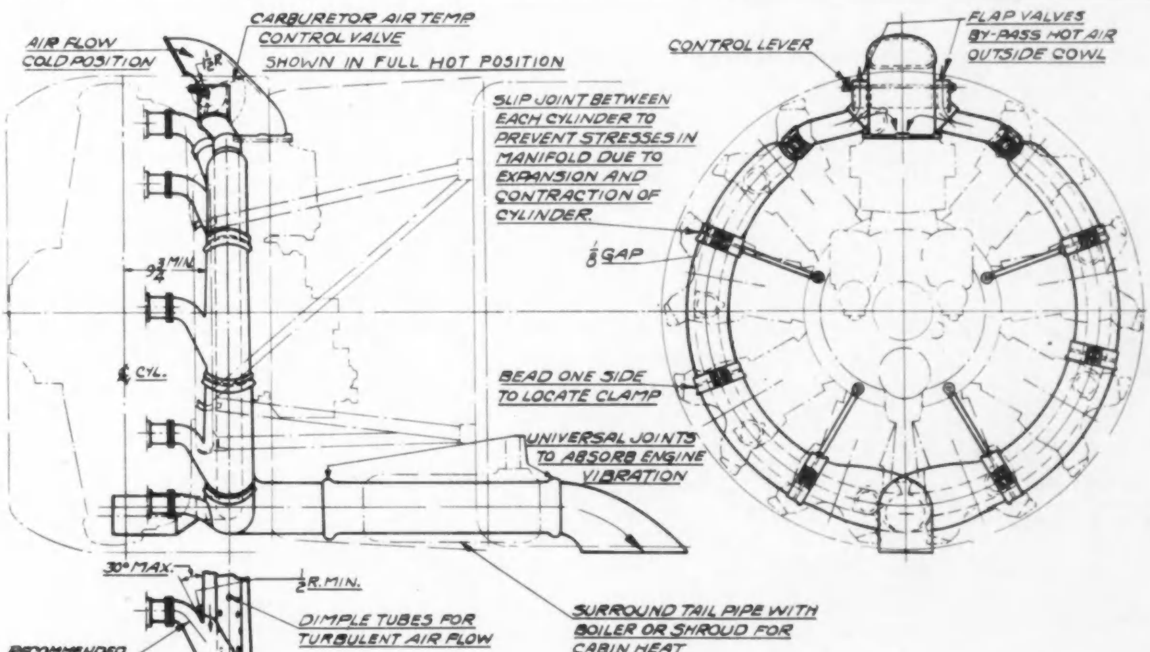



Fig. 12 – Typical Carburetor - Air Heater

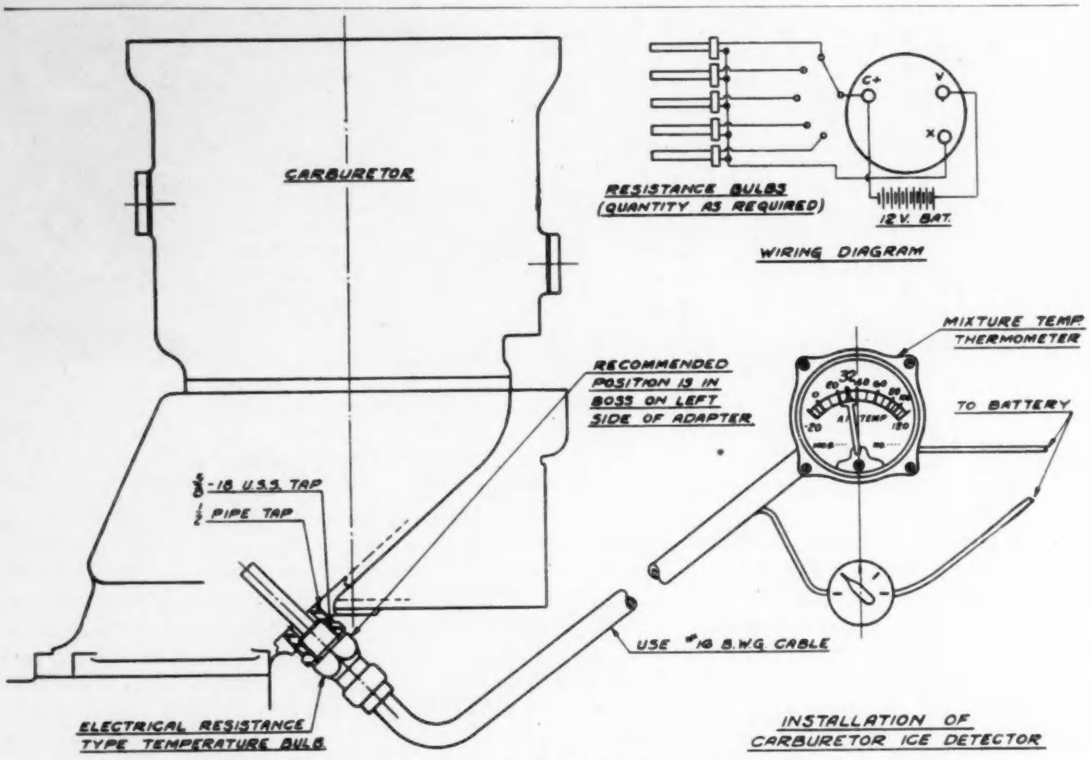



Fig. 13 (*above*) – Fuel-Air Mixture Temperature Indicator

should join the airplane structure near the vertical and horizontal centerline of the engine so as to transmit minimum vibration to the engine-control units through engine vibration. Push-pull tubular controls mounted in ball bearings are preferable. If flexible controls are used they should be braced securely to prevent deflection and unreliable response at the engine. Suitable throttle stops should be provided with supercharged engines to prevent over-power operation and trouble from this source.

## Accessories

The ever-increasing demand for accessory equipment has made it necessary for the engine manufacturer to provide numerous accessory drives. These drives are of necessity limited in power output. Shear members are specified for each drive, and extreme care should be exercised to insure satisfactory lubrication, drainage, and protection against interference with normal engine operation. The accessory vendor should cooperate closely during the installation of his equipment.

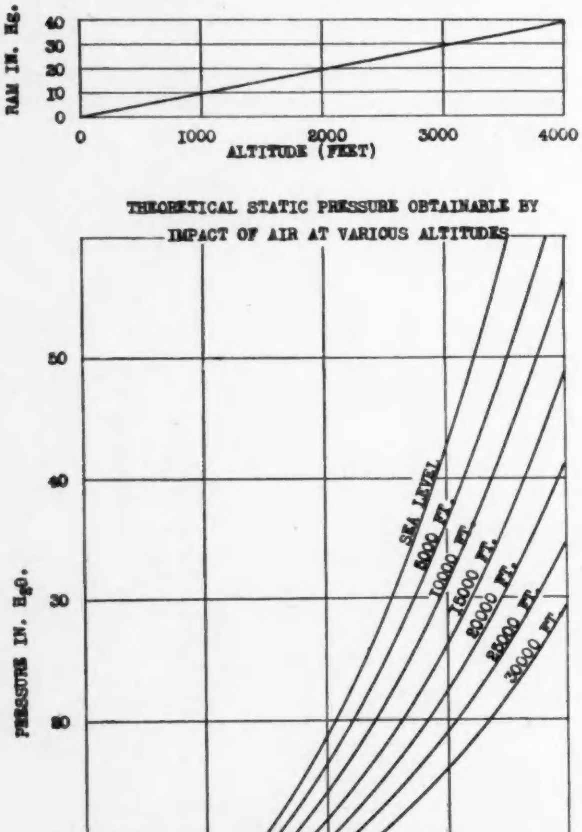



Fig. 14 – Variation in Altitude at Which Rated Power Is Obtained with Change in Ram

## Ground and Flight Tests

Despite careful design, no engine installation can be considered satisfactory until proved so by exhaustive ground and flight tests. One of the major defects of some American aircraft is the inability of the engine to operate satisfactorily during summer operating conditions. It is unfortunate that this condition exists in view of the fact that a thorough initial check of the engine operation requires comparatively little time and expense. Test equipment is as follows, the asterisk indicating the instruments that are standard equipment in most airplanes:

(1) *Tachometer (r.p.m.)
(2) *Supercharger-pressure gage (in. hg.)
(3) *Oil-pressure gage (lb. per sq. in.)
(4) *Oil-inlet thermometer (deg. fahr.)
(5) *Fuel-pressure gage (lb. per sq. in.)
(6) *Carburetor-outlet-mixture thermometer (deg. fahr.)
(7) *Cylinder-head rear spark-plug and cylinder-base thermocouple (deg. fahr.)
(8) *Atmospheric temperature (deg. fahr.)
(9) Oil-outlet thermometer (deg. fahr.)
(10) Carburetor-air-inlet thermometer (deg. fahr.)
(11) Carburetor-air-inlet pressure (in. $H_2O$)
(12) Exhaust-collector back-pressure (in. $H_2O$)
(13) Static pressure fore and aft of cylinder baffles (in. $H_2O$)
(14) Static pressure fore and aft of oil cooler (in. $H_2O$)
(15) Engine rear-compartment temperature (deg. fahr.)

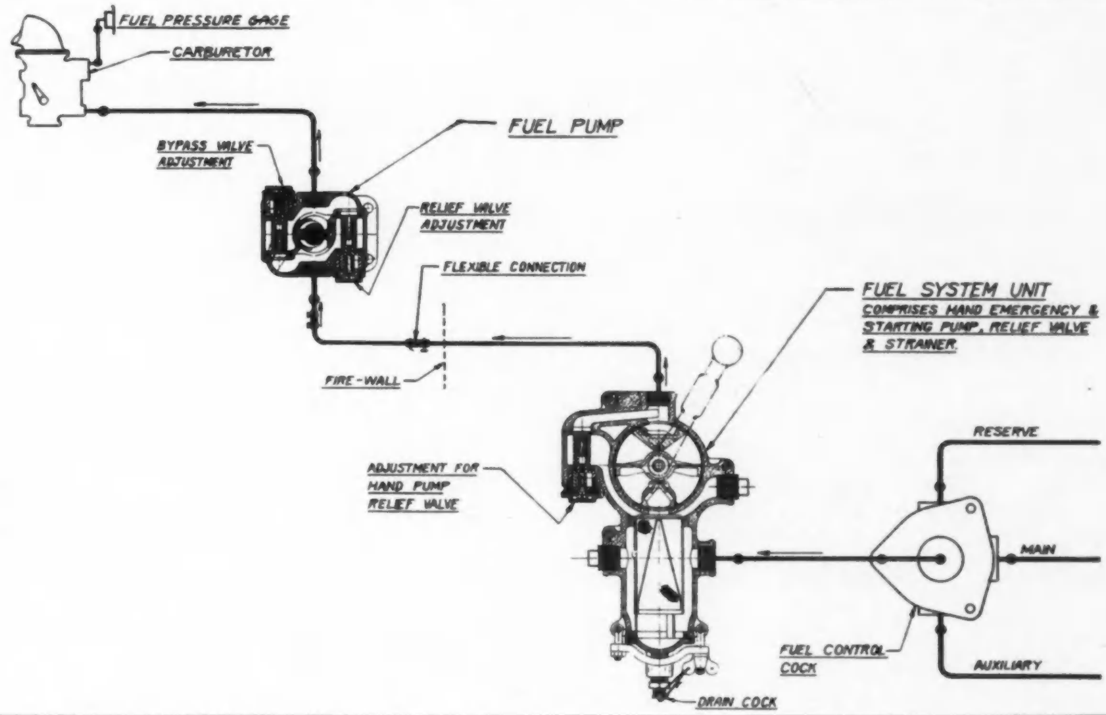



**Fig. 15 – Typical Single - Engine Fuel-System Installation – Internal Relief**

(16) Spark-plug elbow temperature (deg. fahr.)

(17) Magneto cover or coil temperature (deg. fahr.)

(18) Fuel-line temperature (at carburetor inlet) (deg. fahr.)

Note: All static pressure gages should be exposed to the static side of the air-speed pitot.

*Item 9*, the oil-outlet thermometer, is not installed except when oil-cooling troubles are encountered. *Item 11*, for carburetor-inlet pressure, is installed to determine ram and altitude engine performance. *Item 12* is installed to measure exhaust back-pressure, the average of which will determine power loss in accordance with Fig. 3. *Items 13* and *14* will determine cooling-air flow at the cylinders and oil cooler respectively. The other items are self explanatory. Fig. 17 shows the recommended installation of the various pressure measurements. Careful calibration of all instruments before testing the engine is imperative and may prevent expensive alterations made to cure an unsatisfactory condition which actually did not exist.

After checking the engine in accordance with the instructions accompanying it, the ground run should be conducted by operating at 600, 800, 1000 and 1200 r.p.m., provided

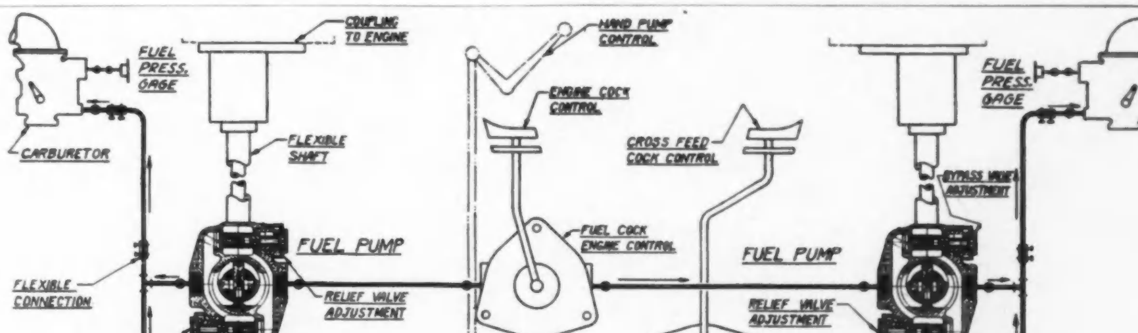



**Fig. 16 – Typical Twin-Engine Fuel-System Installation**

Remote-driven fuel pumps, cross feed, single hand starting and emergency pump.

*Figs. 15 and 16 are reproduced by courtesy of Romec Pump Co., Elyria, Ohio.*

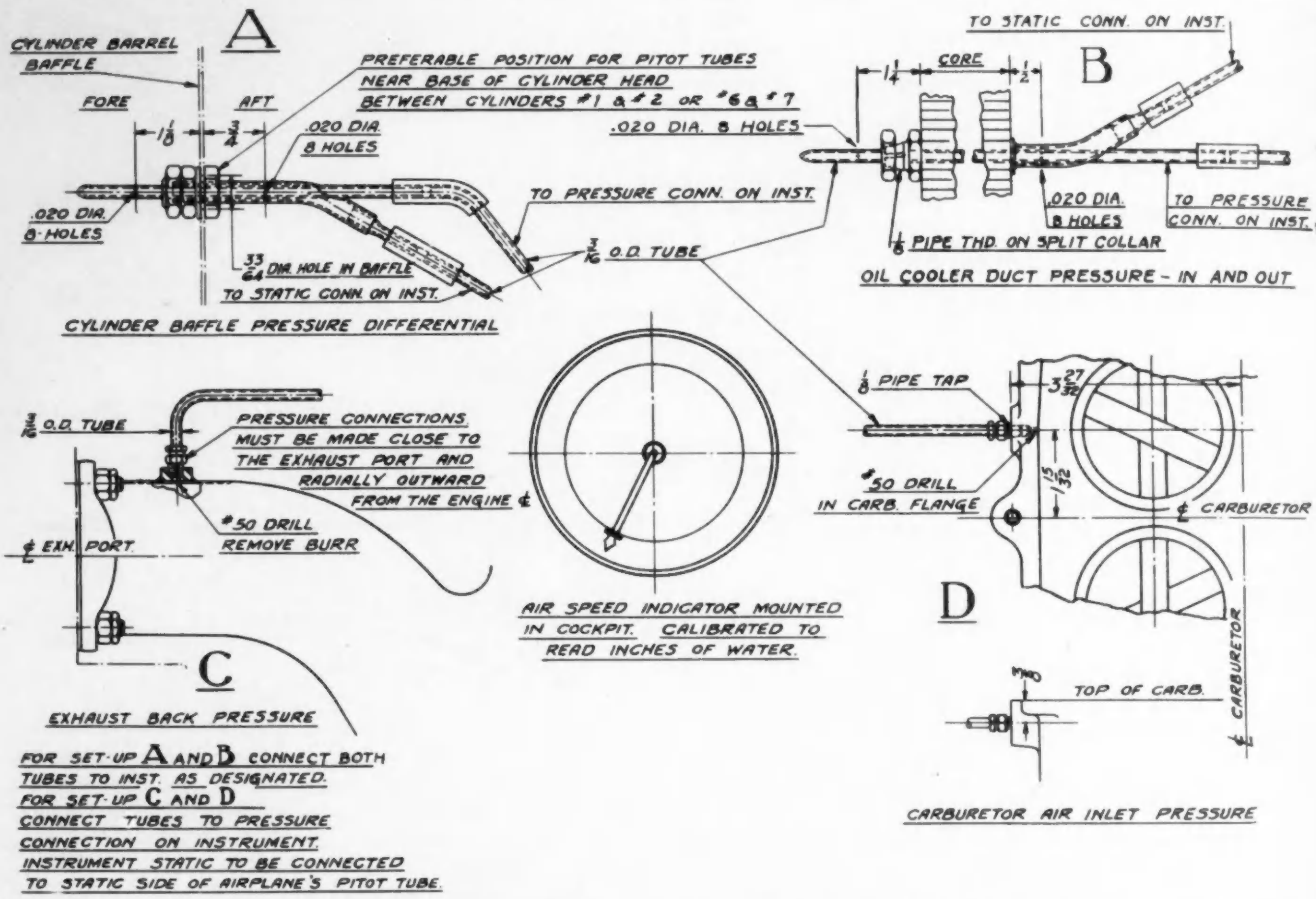



Fig. 17 – Recommended Installation of the Various Pressure Measurements

maximum allowable cylinder and/or oil temperatures are not exceeded. The flight test should be made with gross loading using full take-off power and rated power for climb. The climb should be made at optimum speed, best-power-mixture setting, and should be continued until all temperatures stabilize or exceed the maximum allowable. Level-flight tests should be made with rated and cruising horsepower, checking all readings including the carburetor-air heater. When testing multimotored airplanes, the powerplant should be tested for the most severe flight attitude intended.

The air-temperature limitation suggested for the temperature survey should be 40 deg. fahr. above standard air temperature. At sea level this value represents 100 deg. fahr. In making corrections to observed engine temperature to simulate engine temperature under extreme air conditions, the table of correction factors opposite is recommended.

Taking the difference between the limiting air temperature and the observed air temperature at the altitude under consideration, multiplying this difference by the correction factor, and adding the result to the observed engine temperature gives the corresponding extreme engine temperature to be expected.

These correction factors, while slightly larger than others which have been advocated, are influenced by the fact that the mass flow of cooling air through the cooling passages declines as its temperature increases for any given indicated air speed of operation as explained in Kenneth Campbell's paper[2]

[2] See S.A.E. Transactions, November, 1935, pp. 401-411; "Evaluation of Variables Influencing Air Cooling of Engines," by Kenneth Campbell.

| Item | Maximum Allowable Temperature | Correction Factor |
|---|---|---|
| Cylinder heads | Specified by engine manufacturer | 1.0 |
| Cylinder bases | Specified by engine manufacturer | 0.7 |
| Oil | Specified by engine manufacturer | 1.0 |
| Spark-plug elbow | 248 deg. fahr. (120 deg. cent.) | 1.0 |
| Magneto coil | 175 deg. fahr. (79 deg. cent.) | 0.6 |
| Magneto-coil cover | 160 deg. fahr. (71 deg. cent.) | 0.6 |
| Fuel line | 140 deg. fahr. (60 deg. cent.) | 1.0 |

presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 17, 1935.

## Erratum

In S. D. Heron's discussion of "Future Possibilities of 100-Octane Aircraft-Engine Fuel," by Lieut. F. D. Klein in the August, 1936, issue of the S.A.E. Journal, the first sentence on page 311 should have read: The low-octane number of di-ethyl ether suggests the use of high-molecular-weight, straight-chain ethers for specially low-octane number or high-*cetane* number reference fuels.

# The Correlation of Car and Fuel Vapor-Locking Characteristics

By E. M. Barber and B. A. Kulason
*The Texas Co.*

**IT is the purpose of this paper to present a chart by means of which the vapor-locking characteristics of a gasoline (represented by a curve showing the quantity of vapor formed as a function of the temperature) can be estimated with moderate accuracy for gasolines in the current commercial distillation ranges from the conventional Reid vapor pressure and A.S.T.M. distillation tests on the gasoline.**

**Interpretation and consolidation of car data are facilitated by means of the chart and, in this respect, vapor-lock test data are given for eight 1934, eleven 1935, and several 1936 model cars.**

**The use of the chart and car data is illustrated by a group of sample problems which are specially designed to show the degree of assurance that may be placed on the use of either Reid vapor pressure or A.S.T.M. 10 per cent point alone as a criterion of vapor lock.**

**The problem of evaporation losses from the fuel system, which can be roughly treated by means of the chart, is also discussed briefly.**

**Finally, the application of the chart to the proposed test methods of the C.F.R. and A.P.I. Committees is indicated.**

DESPITE the fact that vapor lock seldom occurs in current-model cars, vapor lock is still an acute problem to the motoring public and, therefore, to the gasoline and automotive industries. Its importance is due not to the few cases that do occur but rather to the methods that are used to prevent its too-frequent occurrence.

The present curb on the occurrence of vapor lock has been obtained by building car fuel systems so that they will vent-off the vapor that the gasoline forms when it is heated to the temperature existing in them, and by manufacturing gasoline to meet compromise front-end volatility specifications. However, every bit of gasoline vapor that is vented out of the fuel system represents a waste of fuel which, in some cars, amounts to as much as 10 or 20 per cent of the fuel supplied to the car. Furthermore, the forced compromise on front-end-volatility characteristics causes a waste and diversion of much of the valuable lighter gasoline fractions that are present in the crude but cannot be used in gasoline manufactured to meet the required volatility specifications.

Numerous careful studies of this problem have almost invariably led to the conclusion that excessively high fuel-system temperature in a few cars is the fundamental cause of the difficulty. In some cases the investigators have recommended a reduction of fuel-system temperatures as a panacea for the problem and, in other cases, they have recommended a reduction in fuel volatility. Both viewpoints deserve consideration, and compromise must be made from both sides. However, it is evident from the conflicting views of the parties involved that a common basis for discussion of the problem is lacking and, therefore, it is difficult to obtain a satisfactory solution.

It is, therefore, the purpose of this paper to present and illustrate the use of a Vapor-Lock Chart which, it is hoped, may serve as a suitable common basis for compromise. By means of this chart it is readily possible to specify the fuel-system temperatures required to prevent both vapor lock and excessive fuel losses when using a gasoline for which the volatility characteristics are known in terms of the Reid vapor pressure and the initial range of the A.S.T.M. distillation. Conversely, it is possible to specify the fuel for a car or group of cars of known vapor-locking characteristics.

### Historical Development

In the vacuum-pump fuel systems of the cars of a few years ago no venting was possible, and vapor lock would occur as soon as a very small amount of vapor was formed. Consequently, it was customary to relate the occurrence of vapor lock to some characteristic of the gasoline that would indicate the formation of a relatively small quantity of vapor. The Reid vapor-pressure test, which indicates the conditions necessary for a gasoline to form four times as much vapor as liquid, was therefore used to indicate the vapor-locking tendency of fuels. When using Reid vapor pressure for this purpose, it was customary to describe the vapor-locking ten-

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 2, 1936.]

dency of cars in terms of the Reid vapor pressure that would just give freedom from vapor lock.

The adoption of mechanical fuel pumps and recent carburetor changes have, however, tended to make fuel systems capable of allowing considerable quantities of fuel vapor to form before it disturbs the flow of liquid gasoline enough to interrupt normal engine operation. Thus, the occurrence of vapor lock is delayed until a relatively large quantity of vapor has been boiled off the gasoline passing through the fuel system. Measurements[1,2] of the limiting quantity of vapor above which vapor lock will occur have shown that, in current-model cars, it varies from as little as 1 or 2 volumes to as much as 50 or 60 volumes[3] of vapor, and that the variation apparently is entirely dependent on the design and installation of the fuel system.

As a result of this increased vapor-handling capacity, which varies over a wide range of values, the early correlations with Reid vapor pressure are no longer adequate to predict the occurrence of vapor lock in current-model cars. The Vapor-Lock Chart has resulted from an attempt to take cognizance of this changed situation.

### Basis for Development of Chart

Before introducing the chart it appears desirable to explain the ideas on which it is based and to define the terms and quantities used.

The following simplified[4], but withal workable, definition of vapor lock has been used as a basis for the construction of the chart and must, therefore, be adhered to fairly rigorously in its application:

*A car will vapor lock if it heats the gasoline in its fuel system to such a temperature that, at the existing pressure, the gasoline will form sufficient vapor to disturb the flow of liquid gasoline enough to interrupt normal engine operation.*

From this definition and the foregoing introductory discussion it is evident that the major factors involved in the problem are:

(1) The tendency of the gasoline to boil and form vapor in the fuel-supply and metering system.

This factor is referred to as the gasoline vapor-forming or vapor-locking characteristic and is described by a curve showing the quantity of vapor formed by the gasoline (expressed in terms of vapor-to-liquid volume ratio) plotted as a function of the temperature (See Fig. 1).

(2) The temperatures and pressures to which the gasoline is subjected in the fuel-supply and metering system.

The temperature of primary importance is that measured at the point of vapor lock.

(3) The vapor tolerance of the fuel system.

This factor specifies the maximum volume of vapor that can be boiled off the gasoline in passing through the fuel system without interrupting normal engine operation. It is expressed in terms of the vapor-to-liquid volume ratio ($V/L$).

Thus, car vapor-locking characteristics are specified by a temperature and a volume measurement, and fuel vapor-

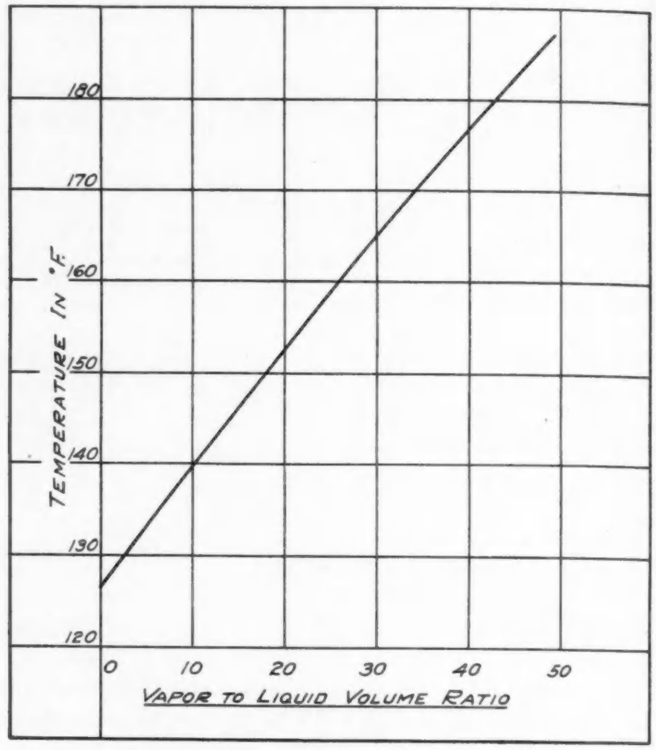



**Fig. 1 – Vapor-Locking Characteristics of a Typical Commercial Gasoline**

Reid vapor pressure, 8.5 lb. per sq. in.
A.S.T.M. 10 per cent evaporated temperature, 135 deg. fahr.
A.S.T.M. 20 per cent evaporated temperature, 161 deg. fahr.

locking characteristics by a curve showing the quantity of vapor as a function of the temperature.

The manner in which the foregoing three factors are brought together to form a workable solution of the problem can be illustrated by the following example:

Determine the fuel-system temperatures at the point of vapor lock in three cars having vapor tolerances of $V/L =$ 10, 25, and 40, when a fuel having the vapor-forming characteristics shown in Fig. 1 is used.

*Solution.* – From the curve of Fig. 1, which gives the vapor-forming tendency of a representative commercial summer-grade gasoline, it is readily seen that the limiting quantities of vapor will form and, according to our definition, vapor lock will therefore occur at fuel-system temperatures of 139 deg. fahr., 159 deg. fahr., and 177 deg. fahr., respectively. To complete this picture, it is only necessary to know the operating conditions that will produce these fuel-system temperatures.

It is to be emphasized that this method gives results which are in good agreement with experiments. Bridgeman reports a deviation of only a few degrees between observed vapor-locking temperatures and temperatures calculated by essentially the same method, and our own tests show an average deviation of only about $\pm 3$ deg. fahr.[2]

### Measurement of Vapor-Locking Characteristics

*Gasoline Vapor-Locking Characteristics.* – Special tests and data not readily available on most commercial gasolines are required normally to establish the vapor-forming characteristics of a gasoline. The inconvenience of obtaining reasonably accurate data on this characteristic has been one of the most troublesome details of the problem.

[1] See S.A.E. Transactions, April, 1933, pp. 157-163; "Vapor-Handling Capacity of Automobile Fuel Systems", by O. C. Bridgeman, H. S. White, and F. B. Gary.
[2] See S.A.E. Transactions, July, 1935, pp. 237-246; "Vapor-Locking Traits of Cars and Their Limitations on Gasoline Volatility", by Neill MacCoull and E. M. Barber.
[3] The term "volume of vapor" used in this paper designates the quantity of vapor equivalent to a vapor-to-liquid volume ratio $V/L$ of 1:1.
[4] Of course, some exceptions to which this simplified definition will not be applicable probably will arise. However, so long as these exceptions are not too frequent, it appears most satisfactory to adhere to some general rule of this sort.

In order to overcome this difficulty Fig. 2 has been developed from a large number of measurements in a special variable-vapor-volume barometric type of vapor-pressure apparatus[5] that is well adapted to determining the vapor-forming characteristics of gasolines.

By means of Fig. 2 the vapor-forming characteristics of a gasoline can be approximated with sufficient accuracy for most practical problems from the Reid vapor pressure and A.S.T.M. distillation tests on the gasoline.[6]

For the sake of simplification, Fig. 2 has been worked out on the basis of 14.7 lb. per sq. in. atmospheric pressure, and the curves of Fig. 3 have been prepared from the perfect-gas laws and the known pressure-temperature relations for hydrocarbons to give approximate corrections[7] for the effect of different pressures on both the boiling temperature and vapor-to-liquid volume ratio. Note that the distillation values always should be referred to a standard atmosphere and that Fig. 3 can be used to make a fairly accurate correction to the A.S.T.M. distillation test.

*Car Vapor-Locking Characteristics.* – Car vapor-locking characteristics can be readily determined by methods which have been described in detail in other papers.[1,2] Briefly, the general method is this:

The vapor-handling capacity is measured by operating the car on a fuel of known vapor-forming characteristics until vapor lock occurs. The temperature of vapor lock is then measured and, from this value and the known vapor-forming characteristics of the fuel, the vapor tolerance ($V/L$) of the fuel system is determined by a process that is the reverse of that described to illustrate Fig. 1. The fuel-system temperatures are then determined at a series of other atmospheric temperature conditions.

Data on the vapor-locking characteristics of current-model cars are not generally available to show the present status of the problem. Therefore, Table 1 has been prepared from the results of tests on a number of 1934-, 1935-, and 1936-model cars in order to show the most pertinent figures. These data were obtained with the cars operating at a speed of 40 m.p.h. and a load equivalent to climbing a 7 per cent grade; data also were obtained when cars were shut down to a 360 r.p.m. idle after the run. In all cases, the vapor tolerance, the point of vapor lock, and the temperature at the point of vapor lock, are given for atmospheric temperatures of 60 deg. fahr., 80 deg. fahr., and 100 deg. fahr.

## The Vapor-Lock Chart

Figs. 2 and 3 and Table 1, together with a brief résumé of the foregoing discussion, are combined on a single sheet designated as the Vapor-Lock Chart. Four sample problems that serve to illustrate the manner in which these data may be used also are given.

By means of the methods and data given on this chart it is possible to determine the car vapor-locking characteristics required to give freedom from vapor lock on any fuel of known vapor pressure and distillation characteristics; and, conversely, it is possible to determine the Reid vapor pressure and A.S.T.M. distillation required for a gasoline to give freedom from vapor lock in any car for which the vapor-locking characteristics are known.

[5] "A Variable-Vapor-Volume Type of Vapor-Pressure Apparatus", by E. M. Barber and A. V. Ritchie, presented at the meeting of the Petroleum Division, American Chemical Society, Kansas City, Mo., April 13-17, 1936.

[6] For illustration see *Problem 1* on the Vapor-Lock Chart.

[7] For illustration of this correction see *Problem 4* on the Vapor-Lock Chart.

## Fuel-System Losses

It is impossible for an appreciable quantity of vapor to be carried through the carburetor jet fast enough to maintain normal engine operation. Therefore, any conception of the problem that postulates the formation of relatively large quantities of vapor in the fuel system before vapor lock occurs must also admit that there will be a loss in fuel economy corresponding to the quantity of fuel vapor that must be vented out of the system.

In a previous paper[2] a number of measurements of such losses were reported and it was shown that losses of 2 per cent to 5 per cent were not uncommon; and losses of as much as 10.5 per cent were measured under fairly normal conditions, although still with no evidence of vapor lock. Not only did the losses cause a decrease in fuel economy, but they also caused a deterioration of 2 to 3 octane numbers in the fuel as supplied to the engine.

It has been found possible to predict approximately (within about 2 per cent loss) the magnitude of the losses that occur in this manner, from the vapor-locking characteristics of the fuel as given by Fig. 2 and a knowledge of the fuel temperature. The method in which this estimation can be carried out is best illustrated by an example:

Determine the fuel-system loss from the car *ALB* (see Table 1) when it is running at 40 m.p.h., 7 per cent grade, in an atmospheric temperature of 80 deg. fahr. on the gasoline specified in Problem 1 on the Vapor-Lock Chart.

*Solution.* – Plot the values defining both the gasoline and car vapor-locking characteristics on Fig. 2. From this plot it will be seen that the car is quite free from vapor lock because, at 146 deg. fahr. (the fuel-system temperature of the car) a quantity of vapor equivalent to only $V/L = 14$ is formed by this gasoline; whereas, the vapor tolerance of the car is $V/L = 31$. However, the 14 volumes of vapor that do form represent (from Fig. 2) a loss of about 7 per cent.

It should be evident from this example that car fuel systems having the combination of a high fuel-system temperature and a large vapor tolerance will be reasonably free from vapor lock, but that they are liable to considerable loss in fuel economy as the result of gasoline boiling in the fuel system.

This waste of fuel would be negligible if it occurred over a very small range of operating conditions and served only as an extra protection against vapor lock. Fortunately, such is the case in a good many cars but, of the 22 models listed in Table 1, there are 5 or 6 having such high fuel-system temperatures that the losses occur over a rather wide range of operating temperatures. Fig. 4 shows the fuel waste for the average and worse-than-average cars of Table 1 when they are run at 40 m.p.h., 7 per cent grade, with various atmospheric temperatures on a fuel having an 8.5 lb. per sq. in. Reid vapor pressure and an A.S.T.M. distillation of:

| | |
|---|---|
| 5 per cent evaporated | 122 deg. fahr. |
| 10 per cent evaporated | 140 deg. fahr. |
| 15 per cent evaporated | 150 deg. fahr. |
| 20 per cent evaporated | 160 deg. fahr. |

which may be regarded as a fairly normal summer-grade gasoline.

It will be noted from Fig. 4 that the losses are limited only by the temperature at which vapor lock will occur and that they are of sufficient magnitude in the worst cars, even at normal atmospheric temperatures, to warrant serious consideration.

*(Text continued on page 355)*

## The Vapor-Lock Chart

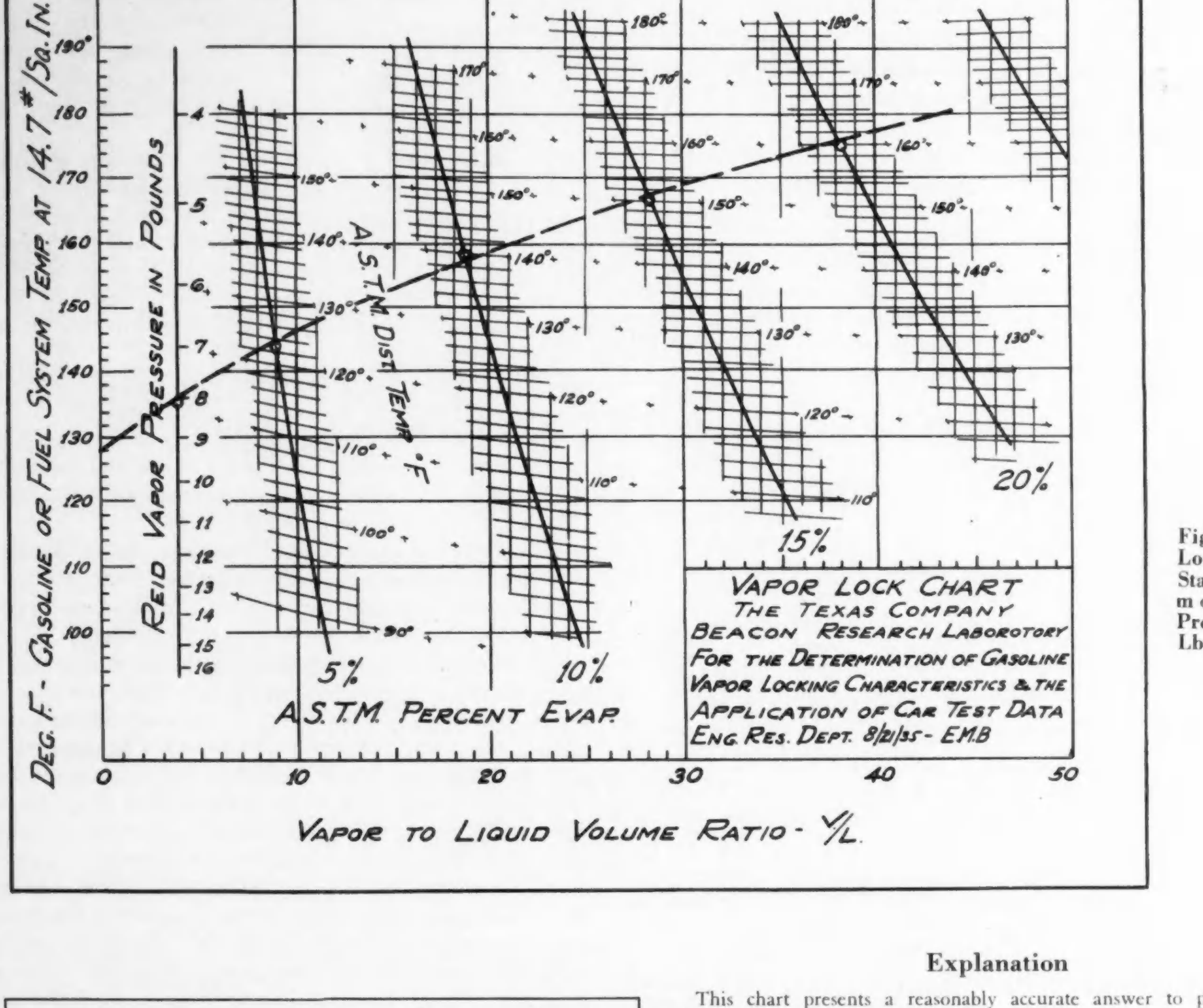



Fig. 2 – Vapor-Lock Chart for Standard Atmospheric Pressure (14.7 Lb. per Sq. In.)

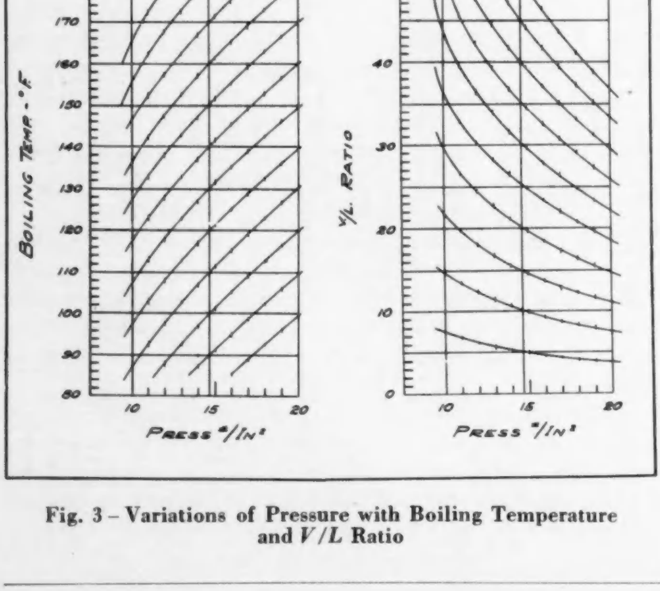



**Fig. 3 – Variations of Pressure with Boiling Temperature and $V/L$ Ratio**

### Explanation

This chart presents a reasonably accurate answer to practically all the questions that are likely to arise in connection with the vapor-lock problem.

*Definition of Vapor Lock.* – A car will vapor lock if it heats the gasoline in its fuel system to such a temperature that, at the existing pressure, the gasoline will form sufficient vapor to disturb the flow of liquid gasoline enough to interrupt normal engine operation.

*Factors in Problem.* – The tendency of the gasoline to boil or form vapor when it is subjected to temperatures and pressures comparable to those existing in the fuel supply and metering system will be referred to as the "Gasoline Vapor-Locking Characteristic" and will be represented by a curve showing the quantity of vapor formed (expressed in terms of the vapor-to-liquid volume ratio, or $V/L$), plotted as a function of the temperature.

The fuel-system or gasoline temperature will be the temperature of the gasoline in the fuel system measured at the point of vapor lock.

The ability of the fuel system to handle vapor will be referred to as the "Vapor-Handling Capacity" or the "Vapor Tolerance", of the fuel system and will be designated by the term "$V/L$". This figure will represent the maximum quantity of vapor that may exist in the fuel system before vapor lock will occur.

Reid vapor pressure and A.S.T.M. distillation, as used in this chart, have their usual significance and are determined by the standard methods.

Fig. 2 is worked out on the assumption that the distillation test is run at sea level, that is, with 14.7 lb. per sq. in. barometric pressure, and that the vaporization will also occur at 14.7 lb. per sq. in. pressure. The correction curves of Fig. 3 show changes in boiling temperature and $V/L$ that result from the vaporization being carried on at different pressures as the result of changes in altitude, fuel-pump pressure, and so on. This temperature-correction chart may also be used to approximately refer distillation temperature to sea-level conditions.

## The Vapor-Lock Chart

**Table 1 – Vapor-Locking Characteristics of Representative Current-Model Cars**

| Car No. | *Steady Run: 40 M.P.H., 7 Per Cent Grade — Point of Vapor Lock | V/L | Atmospheric Temperature 60 deg. fahr. | 80 deg. fahr. | 100 deg. fahr. | Idle After Steady Run — Point of Vapor Lock | V/L | Atmospheric Temperature: Two-Minute Idle 60 deg. fahr. | 80 deg. fahr. | 100 deg. fahr. | Four-Minute Idle 60 deg. fahr. | 80 deg. fahr. | 100 deg. fahr. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| *1934* | | | | | | | | | | | | | |
| 23 | Fuel pump | 29 | 116 | 129 | 142 | Carburetor | 35 | 123 | 135 | 149 | 129 | 140 | 155 |
| *WGC* | Fuel pump | 14 | 135 | 145 | 155 | Fuel pump | 19 | 163 | 175 | 189 | 165 | 179 | 185 |
| 25 | Fuel pump | 13 | 100 | 116 | 133 | Carburetor | 6 | 106 | 122 | 137 | 112 | 129 | 144 |
| 24 | Fuel pump | 12 | 110 | 123 | 136 | Carburetor | | 125 | 138 | 152 | 130 | 144 | 157 |
| 21 | Fuel pump | 28 | 103 | 116 | 130 | Carburetor | $>50$ | 127 | 141 | 155 | 144 | 160 | 174 |
| 20 | Fuel pump | 13 | 103 | 120 | 136 | Carburetor | 43 | 134 | 151 | 166 | 139 | 146 | 167 |
| 26 | Fuel pump | 34 | 113 | 127 | 140 | Carburetor | $>50$ | 120 | 132 | 149 | 125 | 138 | 155 |
| 22 | Fuel pump | 11 | 90 | 105 | 118 | Carburetor | 21 | 115 | 127 | 141 | 125 | 137 | 149 |
| *1935* | | | | | | | | | | | | | |
| 28 | Fuel pump | 20 | 105 | 119 | 133 | Carburetor | 25 | 110 | 125 | 140 | ... | ... | ... |
| 29 | Fuel pump | 48 | 149 | 163 | 176 | Fuel pump | $>50$ | 170 | 186 | 202 | 174 | 191 | 208 |
| *GK* | Fuel pump | 23 | 128 | 145 | 162 | Carburetor | 43 | 143 | 160 | 177 | 151 | 168 | 185 |
| 30 | Fuel pump | 18 | 98 | 114 | 130 | Carburetor | $>50$ | 131 | 147 | 163 | 137 | 152 | 167 |
| 32 | Fuel pump | 22 | 104 | 118 | 132 | Carburetor | 37 | 129 | 147 | 164 | 140 | 157 | 174 |
| 31 | Fuel pump | 12 | 117 | 134 | 152 | Carburetor | 25 | 133 | 150 | 167 | 142 | 159 | 177 |
| *LFM* | Fuel pump | 25 | 129 | 142 | 156 | Fuel pump | 26 | 138 | 159 | 175 | 139 | 160 | 176 |
| *ALB* | Fuel pump | 31 | 128 | 146 | 163 | Carburetor | $>50$ | 138 | 159 | 180 | 140 | 166 | 190 |
| 34 | Fuel pump | 26 | 112 | 125 | 138 | Fuel pump | 26 | 124 | 139 | 154 | 132 | 143 | 163 |
| 33 | Fuel pump | 15 | 93 | 109 | 126 | Carburetor | $>50$ | 145 | 158 | 170 | 150 | 163 | 176 |
| 35 | Fuel pump | 16 | 130 | 144 | 158 | Fuel pump | 28 | 140 | 153 | 167 | ... | ... | ... |
| *1936* | | | | | | | | | | | | | |
| 39 | Fuel pump | 21 | 100 | 118 | 136 | Carburetor | $>50$ | 122 | 139 | 156 | 129 | 147 | 165 |
| 38 | Fuel pump | 22 | 101 | 118 | 135 | Carburetor | 48 | 142 | 164 | 187 | 154 | 176 | 198 |
| 37 | Fuel pump | 16 | 111 | 124 | 137 | Carburetor | 23 | 135 | 148 | 162 | 150 | 160 | 171 |

*Under other operating conditions it is to be expected that the vapor-handling capacity of the car will remain substantially constant. Fuel-system temperatures, however, vary considerably, the average result of tests on several cars indicating an increase of about 1 deg. fahr. per 1 per cent grade increase over the range from level road to maximum load. Thus, to correct these values to level-road conditions, subtract 7 deg. fahr. from the fuel-system temperatures. Over the range of 30 to 50 m.p.h. the average of several tests showed no appreciable effect on fuel-system temperatures but, on increasing the speed to 60 m.p.h., the temperatures increased 7 deg. fahr., so that 60 m.p.h., level road, is approximately equal to 40 m.p.h., 7 per cent grade. Thus the data of Table 1 can be approximately corrected to other operating conditions.

### Examples

The following sample problems illustrate the use of the chart and data:

*Problem 1.* – Determine the vapor-locking characteristics of a gasoline having the following Reid vapor pressure and A.S.T.M. distillation: Reid vapor pressure, 8 lb. per sq. in.; 5 per cent evaporated point, 122 deg. fahr.; 10 per cent evaporated point, 140 deg. fahr.; 15 per cent evaporated point, 150 deg. fahr.; 20 per cent evaporated point, 160 deg. fahr.

*Solution.* – Plot these tests at the proper points on the lines designated for them in Fig. 2 and connect the points with a smooth line. The vapor-locking characteristic of the gasoline is then read directly from the ordinate and abscissa of the chart (See circles and dashed line on Fig. 2).

*Problem 2.* – Determine the temperature at which the car *ALB* ($V/L = 31$) will vapor lock on the gasoline of Problem 1.

*Solution.* – Enter the Fig. 2 at the abscissa $V/L = 31$, and go vertically to the line representing the gasoline characteristics, and thence horizontally to the left-hand temperature scale at 168 deg. fahr. This value means that this car will vapor lock on this fuel when the fuel temperature reaches 168 deg. fahr. From the relationship between the fuel-system and atmospheric temperatures this figure may be translated to the atmospheric temperature at which vapor lock will occur at 40 m.p.h., 7 per cent grade – from Table 1, about 106 deg. fahr.

*Problem 3.* – Determine the gasoline characteristics required to give protection from vapor lock to the following four cars when they are run at 40 m.p.h., 7 per cent grade, 100 deg. fahr. atmospheric temperature: Car No. 30 ($V/L = 18$); Car No. 32 ($V/L = 22$); Car No. 34 ($V/L = 26$); and Car No. 33 ($V/L = 15$).

*Solution.* – Plot the $V/L$ value for the car against the fuel-system temperature measured under the desired operating conditions. A gasoline having vapor-locking characteristics that will just give protection to these cars, under the prescribed conditions, will be defined by a line passing just above the plotted points. This gasoline must have an A.S.T.M. 5 per cent evaporated point of 104 deg. fahr. or greater; a 10 per cent evaporated point of 120 deg. fahr. or greater; and a 15 per cent evaporated point of 128 deg. fahr. or greater.

*Problem 4.* – At an atmospheric pressure of 14.7 lb. per sq. in. a certain gasoline will form 30 volumes of vapor per volume of liquid at 150 deg. fahr. Determine the $V/L$ and temperature corresponding to a barometric pressure of 12 lb. per sq. in.

*Solution.* – Enter the temperature scale of Fig. 3 at $T = 150$ deg. fahr., and cross to $P = 14.7$ lb. per sq. in.; then follow parallel to the family of diagonal curves to $P = 12$ lb. per sq. in. and return horizontally to the temperature scale at $T = 138$ deg. fahr. Do the same for the $V/L$ value – this gives $V/L = 36$ at $P = 12$ lb. per sq. in.

*(Text continued from page 353)*

It is evident that a solution of the vapor-lock problem requires cooperation between the fuel refiner and the car builder. The fuel refiner should provide a fuel of such characteristics that it will not form more than the allowable quantity of vapor at the fuel-system temperatures existing in current-model cars. On the other hand, the car builder should insure that the fuel system of his car will not run so hot as to require a fuel of high-boiling characteristics that will result in poor starting and warming-up performance, or that will cause the fuel refiner to waste much of the valuable lighter gasoline fractions.

Reduction of the vapor-locking tendency of cars by increasing the vapor-handling capacity rather than by decreasing the fuel-system temperature is seen to produce an unnecessary decrease in fuel economy. These losses actually occur in some of the current-model cars listed in Table 1 and are particularly severe in those cars having high fuel-system temperatures combined with large vapor-handling capacities.

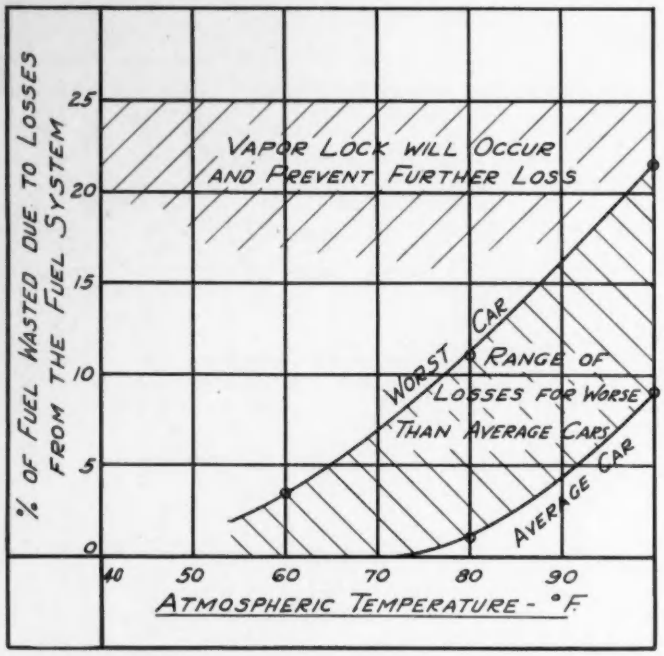



**Fig. 4 – Fuel Waste Vs. Atmospheric Temperature**
In various cars of higher than average fuel-system temperature and vapor-handling capacity (listed in Table 1) when operating with a representative summer-grade gasoline.

The authors gratefully acknowledge use of many ideas which were taken from the Bridgeman reports and papers, often without direct reference to the work.

The authors also express their appreciation for the assistance of C. E. Cummings and Bruce Hegeman in the formulation and development of this work.

# Discussion

## Direct Measurement of *V/L* Ratios Indicates Discrepancies

**—J. M. Campbell, W. G. Lovell, and M. J. Mulligan**

*General Motors Research Laboratories*

THE authors of this paper have presented an excellent discussion of the application of the vapor-to-liquid ratio concept to the vapor-lock problem. In particular the authors have made a serious attempt to correlate the vapor-forming properties of a gasoline with its A.S.T.M. distillation curve by the graphical method shown in Fig. 2 of their paper.

As we understand this chart, the authors intend it to be applicable to a wide variety of gasolines. However, after careful study of this chart, we have found wide discrepancies between vapor-to-liquid ratios estimated from this chart and vapor-to-liquid ratios which are known from direct measurements. This discrepancy can mean either one of two things: first, that there is some fundamental difference between the vapor-to-liquid ratios determined by different laboratory methods of measurement or, second, that the chart may not be applicable to gasolines having widely different distillation characteristics.

One of the first forms of apparatus for the direct measurement of vapor evolution of gasolines was described by G. G. Brown in 1930.[a] This apparatus was designed to give a direct measure of the vapor evolved from a sample of gasoline of known volume. A simplification of this same type of apparatus was described by Francis in 1932.[b] More recently an apparatus similar to Brown's apparatus has been in use at the General Motors Research Laboratories. Fig. *A* shows in general how this apparatus was constructed.

[a] See University of Michigan 1930 Engineering Research Bulletin No. 14.
[b] See *Oil and Gas Journal*, Mar. 17, 1932, p. 22; "Method for Determining Vapor Lock", by C. K. Francis.

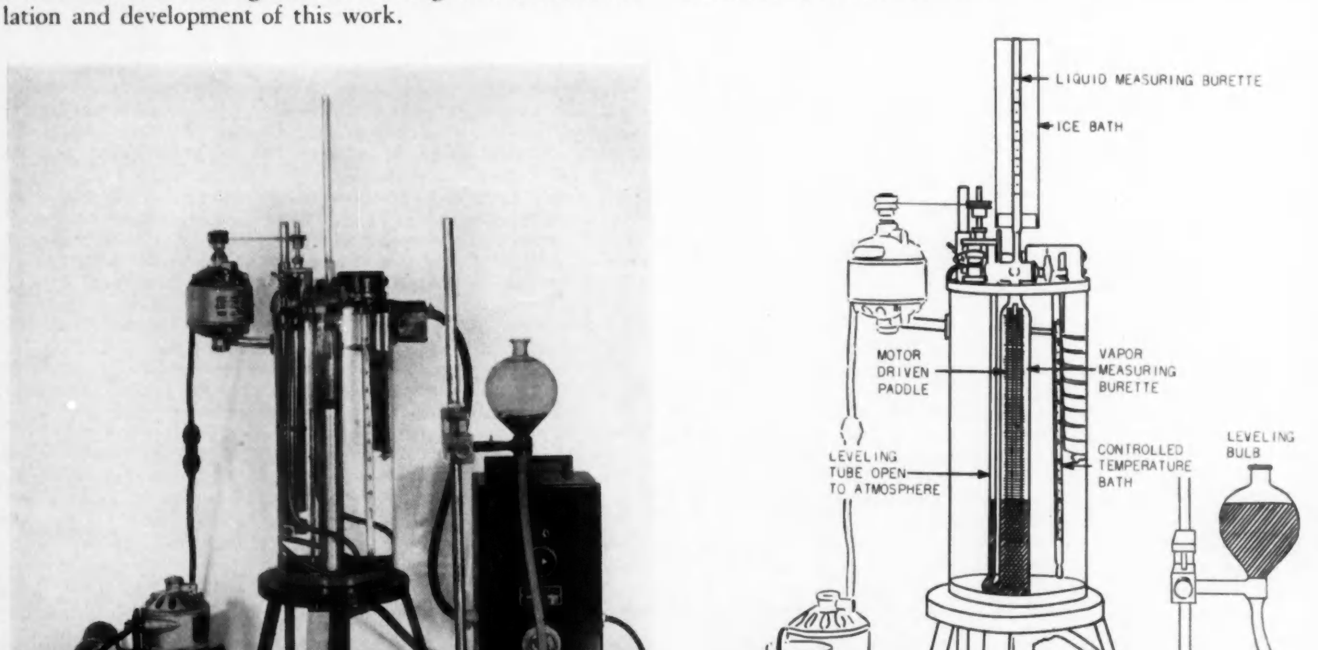



**Fig. *A* (*Barber-Kulason Discussion*) – Apparatus for Direct Measurement of Vapor-to-Liquid Ratios**

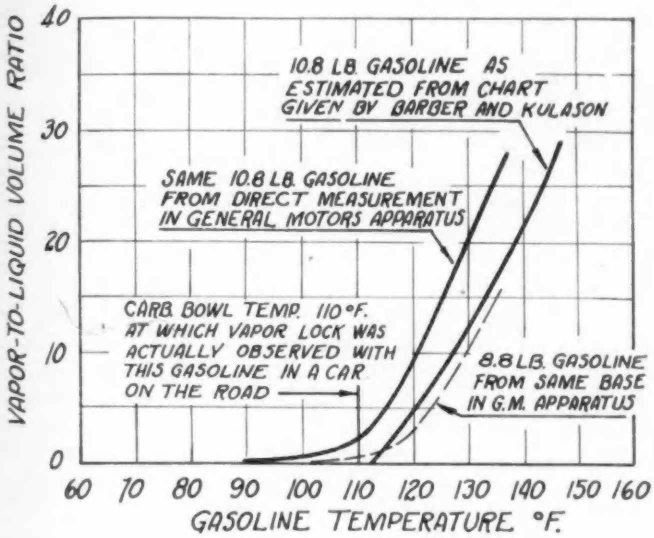



**Fig. B** *(Barber-Kulason Discussion)* – **Comparison of Vapor-to-Liquid Volume Ratios Obtained by Direct Measurement and from Vapor-Lock Chart**

The apparatus consists essentially of a 100-cc. glass burette calibrated to 0.2 cc. and provided with a motor-driven stirrer extending throughout its length. This burette was filled with mercury and connected to the leveling bulb shown in the figure. At each temperature at which measurements were made the pressure of the vapor in the burette was brought to atmospheric by means of the leveling bulb. To facilitate this adjustment, there was provided a small tube open to the atmosphere and shown in the figure adjacent to the burette. The burette also was provided with a three-way stop-cock at the upper end, which enabled liquid samples to be metered into it from a 5-cc. burette immersed in an ice-bath.

In operating this apparatus, the samples of gasoline were cooled and drawn into the measuring burette over dry mercury. The samples were introduced as received, that is to say, they were not dried nor was any attempt made to saturate them with water or air. It was believed that this method represented the nearest approximation to conditions actually existing in the fuel system. Particular precautions were found necessary to insure equilibrium vaporization and, for this reason, the apparatus was provided with a motor-driven stirring mechanism for agitating both the liquid and the gas contained in the measuring burette. From the vapor volumes measured at various temperatures and at atmospheric pressure a curve of vapor-to-liquid ratio against temperature could be drawn.

Fig. *B* shows a comparison of some direct measurements of vapor-to-liquid ratios made in this apparatus with vapor-to-liquid ratios which have been estimated by the use of the chart presented by Barber and Kulason. The gasoline represented in this figure was a sample taken during some road tests where vapor lock was being observed. It had a Reid vapor pressure of 10.8 lb. per sq. in., an initial point of 108 deg. fahr., 10 per cent distilled at 125 deg. fahr., and 20 per cent distilled at 150 deg. fahr. with a 2 per cent distillation loss. Fig. *B* shows that there is a considerable discrepancy between the vapor-to-liquid ratios observed by direct measurement in this apparatus and those estimated from the chart. The position of the 8.8 lb. per sq. in. gasoline indicates that the discrepancy amounts to the equivalent of about 1.5 lb. per sq. in. in Reid vapor pressure. The magnitude of this discrepancy is typical of that observed on a number of different gasolines which have been compared by the two methods.

It might be pointed out that vapor lock actually was observed with this 10.8 lb. per sq. in. gasoline in a car at a carburetor-bowl temperature of 110 deg. fahr. This value is about 2 deg. below the temperature at which vapor evolution should begin according to the vapor-lock chart under discussion, if we interpret the chart correctly.

In addition to the lack of agreement between vapor-to-liquid ratios determined from this chart and vapor-to-liquid ratios determined by direct measurement, the values obtained from this chart do not appear to be consistent with values given by Bridgeman for blends of gasoline and butane[e].

That a chart of the type presented by Barber and Kulason in their Fig. 2 may not be applicable to a wide variety of gasolines is indicated by the following data which were obtained in the apparatus which we have described previously:

| Gasoline | Reid Vapor Pressure, lb. per sq. in. | A.S.T.M. 10 Per Cent Evaporation, deg. fahr. | Measured $V/L$ at 124 deg. fahr. |
|---|---|---|---|
| Natural gasoline blend | 12.2 | 110 | 64 |
| Natural gasoline blend | 11.8 | 110 | 57 |
| Butanized commercial gasoline | 13.8 | 109 | 45 |
| Butanized commercial gasoline | 12.0 | 111 | 23 |

The gasolines chosen for this illustration had 10 per cent evaporated points at 110 deg. fahr., within experimental error and, according to the vapor-lock chart, all of them should have vapor-to-liquid ratios of 22 at 124 deg. fahr. Actually the vapor-to-liquid ratios as measured directly in the apparatus described previously varied from 23 to 64.

We therefore feel that, in view of the growing importance of vaporization phenomena in fuel systems, further investigation of laboratory methods of determining vapor-to-liquid ratios may be necessary. Since the concept of the vapor-to-liquid ratio has become a part of our technical vocabulary, it is highly desirable that we have methods of determining this ratio by accepted and reproducible methods.

[e] See S.A.E. Transactions, April, 1933, pp. 157-163; "Vapor-Handling Capacity of Automobile Fuel Systems", by O. C. Bridgeman, H. S. White, and F. B. Gary.

## A New High-Octane Blending Agent

**By H. E. Buc and E. E. Aldrin**

*(Continued from page 340)*

Conclusions drawn so far indicate that:

(1) The 100-octane number isopropyl-ether blend has a minimum specific fuel consumption 13 per cent lower than the 92-octane number regular gasoline when run under cruising conditions.

(2) The 100-octane number isopropyl-ether blend has a minimum specific fuel consumption 5 to 7 per cent higher than the 100-octane number iso-octane blend under cruising conditions. This result is in proportion to its lower heat content.

(3) The lower economy of isopropyl ether may be overcome by going above 100-octane number which is possible with this material.

(4) The 100-octane number blends of isopropyl ether and iso-octane are equal in power output and consumption under high power conditions such as are used during take-off.

## Discussion

# Advantages of Iso-Pentane in Blends Pointed Out

**—J. H. Doolittle**
***Shell Petroleum Corp.***

ISO-OCTANE, due to its lower volatility, will permit the use of a larger percentage of iso-pentane in a blend of iso-octane (or isopropyl ether), iso-pentane, and aviation base stock.

Iso-pentane actually costs no more to manufacture than does 74-octane aviation base stock and, in the required amounts, it is more readily available. Its octane number is 88 against 74 for the best available aviation base stock. It has a very high lead susceptibility. It may be seen readily then that the use of iso-pentane would make the iso-octane appear in a more favorable light.

The use of iso-pentane also would improve the distillation curve, and a blend of iso-octane, iso-pentane, and aviation base stock can be made that will give almost the same distillation curve as typical aviation gasoline. This blend would improve starting characteristics which, from the curve shown in Fig. 3, might be considered bad.

# Body Designing Procedure

By George J. Mercer
*Consulting Body Engineer*

**EVOLUTION of body engineering is recalled with the traditional practices of the profession and the difficulty of obtaining information and instruction. Relations and locations of side-sweep, turnunder sweep, and belt line are discussed; definite suggestions are made and design procedure outlined.**

**How the first visual impression or "eye appeal" of a new design affects public acceptance is emphasized, and the special influence of this factor upon women is pointed out.**

**Responses from three authorities in body design to a twelve-point questionnaire on debatable policies and principles give an indication of modern body-design trends and practice.**

PRIOR to the commencement of work upon any mechanical operation, even of the simplest character, plans in some form are required as a guide to insure that the final results measure up to specifications.

The men who are directly responsible for the making of these plans are usually called engineers. There is a wide variety of businesses wherein engineers function as executives, and so a prefix is added to the title "Engineer" to designate the branch in which he has qualified as an expert.

The title Body Engineer is of very recent origin, and it is really a courtesy title, as there is no college that grants it. In 1909, when I opened an office in New York City as a consultant, designer, and draftsman to the body trade, I was at a loss what to name my business, what to put on the stationery. After talking it over with several people, I adopted the suggestion of one and used "Automobile Body Architect". The name engineer I had not then heard applied to body men; it came into use at a much later date.

Every form of engineering work, in fact, every form of business, has some peculiarities in connection with its practice that are comparable with the idioms in a language. These peculiarities are recognized as standards by those who use them but, to men in other lines of work, they are the subject of criticism. I have in mind two that are the most outstanding in body engineering: one is the custom of drawing the plan view on top of the elevation in the working draft, and the other is disregarding the use of the third-angle projection on these same large body drawings. It is quite common to see both first- and third-angle projection used on the same drawing.

I recall the older men, when I first went to work, speaking of the geometrical formula used for developing the lines and surfaces on the body draft as the "French rule". The French are given credit as the first to correlate or group the various geometrical rules that had come down through the years in coach building; to these they added some from the boat-building industry; and the whole became known as the French rule.

These old formulas constitute the base of what we use for development in body work today; some new ones have been added to meet different development needs, and some of the old have been dropped.

In addition to the necessary instructions as to size, and so on, that the body-layout man has at the outset of his work of making a body draft, there are the two major curve values known as the side sweep and the turnunder sweep. Both these lines are straight lines in the elevation; the side sweep is a horizontal line in the elevation at the height of the belt line which is the lower edge of the belt molding. It extends from front to rear on the side and, as it extends around the corner and the rear, it is called the rear sweep. In the plan it shows as a curved line. Other side-sweep lines are drawn at different heights, but they are developed from the one major line determined at the start of the drawing.

The side-sweep line is not always determined at the belt line; near by places to the true belt line are used and this practice is permissible, but there is a common error when another place than the true belt line is used to locate the side sweep and this new location is marked on the drawing as "belt line". I have suggested at times, and repeat now, that it would be an advantage to use the names employed by British and French body men, calling the lower edge of the belt molding the "belt line", the lower edge of the window opening the "elbow line", and the space between these two lines, the "waist". The belt molding is the molding below the windows on the side of the body and, when there is more than one molding, the lower one is the belt.

The turnunder sweep is a vertical line in the elevation; it is located customarily at the front edge of the body pillar at the rear of the door opening. If there are two doors on the side of the body, this line will be at the rear of the rear door, and at the rear of the single door on the side when there is only one door. This line shows as curved in the end views. It is always located at the widest part of the body.

These two sweep lines are true continuous curved lines but not true radii and, in order to reproduce them and the lines and surfaces that are developed from them, they are drawn full size; thus patterns and templets made from the drawing are used when making the body. The full-size body draft is a big affair; that is why the plan is drawn over the elevation and, to minimize the number of lines crossing, this plan is generally drawn first-angle projection.

The end views, both front and rear, are drawn half, showing one side from the centerline and, although the elevation

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is drawn showing the left side, the end views show the right half of the body. Here again is a violation of third-angle projection, but fewer lines cross so there is a sensible advantage. At times the elevation is drawn as a centerline drawing, which is a section through the centerline. When the elevation shows this section, the plan and end views become true third-angle projection; the centerline drawing of the elevation will show the right side of the body, and thus the views of the plan and the ends will conform to it.

The procedure at the start of any project is interesting; human nature is inquisitive and likes to know the how and why at the birth of an idea that later becomes a reality. The first step in the making of a body design is a mental picture in the mind of some person, who thus visualizes the possibilities of a new type of body that is a change over the prevailing design. The change naturally must embody some betterment or advance. This betterment will be the selling idea that will attract others and, to accomplish this purpose, a drawing or drawings are made to convey in a practical form the mental picture of one person to other minds. The drawing is the first step in the selling of the mental picture.

There is one angle of major importance that enters into the selection or adoption of any new design, namely, the attitude of the public. After all has been said and done, the public eventually will have to pass upon it – that will be the final and most important test. Let us suppose, for example, that a new-model car has been brought out, and we shall assume that it has all the merits of performance and economy in operation and remains sold 100 per cent once a buy is consummated. Then, suppose we have another car of about the same class and price; we shall assume also that No. 2 car is not as good a buy as No. 1, but it does have the edge in a better body design, or rather a better looking body design. It is safe to bet that the salesman handling No. 2 car will have an easier time than the salesman handling No. 1. Provided that No. 2 should show that its mechanical defects made it a poor buy, the better-looking body could not carry any burden such as that. However, there would be in the minds of many buyers of No. 1 car a lingering wish that conditions were more nearly equal, so that they could buy the other car. They would be buying No. 1 car under pressure, not from choice. This is a fair statement of the eye appeal that is a factor in the selling of cars, just as it is in the selling of furniture, clothes, and so on; it is that something that can never be overlooked.

It is admitted that there is no rule to measure taste or the sense of style in the average person. The why that people choose or show preference cannot be estimated, nor is there any known argument that will make them change their opinion voluntarily. The average person feels competent to pick and choose in the matter of taste without asking advice or guidance and, usually, the less competent they are judged from the viewpoint of those who are conversant with the subject, the more sure-fire are they in making a quick decision. The newsboy on the corner, looking at a new design of motor car, possibly one that may be the product of very competent men and may have cost a lot of money to build, without a moment's hesitation, will say: "It's a peach" or he may say: "It's a wash-out", and he is quite certain that his opinion is the last word in the matter.

When the new design is first presented as a picture to be discussed at the initial showing, it almost seems as if it were surrounded with question marks, and these questions have to be settled to the satisfaction of the best judgment of those interested before further progress can be made. It is a very important moment, this first showing; body designers all recognize how vital is the initial revealing of their efforts. In a conclave with the discussion of the new design as the topic, difference of opinion may develop into a discussion in which the point of view becomes of more importance than the design. A compromise is not a decision; it usually will lean too much to the conservative side, leaving little room for anything new. There is an old saying that too many cooks spoil the broth, and Napoleon is credited with saying that there was one thing worse for an army than having a poor general, and that was for it to have two good ones at the same time.

For nine years I had an office on Broadway, New York City, and made drawings, designs, and acted in a consulting capacity. My customers were body builders, car manufacturers, private owners, and the salesmen from the various agencies along Broadway. I learned through this contact with various people who came into my office that very few people have a definite idea as to what they want in the way of design or style, but they all seemed to have defined ideas of what they did not want or like.

### Women's Opinions Helpful

Another thing that I learned was that women, when of the type who choose for themselves, when ordering a body or selecting a design, were usually more precise and definite in knowing what they wanted than the average man. I always found them capable in expressing their wishes, and their judgment was invariably good. I recall when I worked as a draftsman for Quinby & Co., Newark, N. J., that one of the body designs that was popular had a double molding that I suggested detracted from the appearance, but I was unable, however, to sell the idea of changing. One day a woman customer came in, ordered a body, and selected the type I have mentioned but, pointing to a finished body, she said: "Leave off that second molding on my body". The improvement was so noticeable that ever after that those bodies were made minus the second molding.

It seems necessary to have varied influences injected into the designing of a body to keep the designer thinking in the right direction. I recall a friend who took over the management of a business that is very similar to ours, and his company planned to make the bodies for a small car it proposed to bring out. This friend said to me: "I wish you could recommend a sort of half-cracked body designer – one that has visionary ideas". I looked at him to see if he was serious. He said: "Yes, I mean it; I think I could go places with the right kind of a nut".

Satisfactory results over a period of time are best obtained with practical men in control of the designing of the body. It is true that practical men become conservative, and we know that conservatism is the basis of decay; therefore, a certain amount of the radical and ultra-extreme is injected to keep the thinking actively progressive. Up to the present the majority of designers are men who have come up through the ranks and know shop and, therefore, they have a balanced experience. A designer must be practical; he must be an artist and also a salesman. The future designer will be quite different in makeup from what he has been heretofore; the necessities of the conditions will develop the men to fill these positions. I would like to add that I have tried to interest two colleges, that is, I tried with one and then the other; I suggested that they add "body engineering" to their studies. I received some encouragement and have not given up hope that I shall succeed. If body engineers would sup-

port this project energetically, I feel confident that we could succeed in getting the recognition that body engineering deserves as a right.

Selling the design – this term embraces all the steps from the first showing of the drawings up to the full-size model.

The first stage is the making of the drawings that focus and define the preliminary mind-picture. This mind-picture is a concept of the requirements and needs for a particular type body; therefore, the drawings are usually pictures made within a prescribed mental area that is defined and limited. Designers usually make these first pictures as complete as possible. There is an expression used by artists that a picture has life, and the body designer tries to have his pictures show true by having the drawing a perspective view showing the front. Thus a full concept is conveyed to the mind of the observer and, if the picture is colored, an even better sense of life and fullness is conveyed.

A draftsman unconsciously develops the faculty of seeing even more than the drawing shows; whereas the person that is not used to reading drawings, sees less than the lines laid down. The designer familiar with this fact very wisely presents his pictures of the new design so that questions will not have to be asked. No amount of talking will sell the design like a good picture that talks for itself. Eye appeal is better than ear appeal. Avoiding a discussion is good policy on the part of the designer.

Once the acceptance of the picture has been accomplished, the next stage in order is the small model, either wax, plaster, or wood, and made very exact and the same size as the picture, which is usually one-fourth size and, in order to make a correct model, a drawing with the surfaces and lines developed precedes the actual work in clay. Then, next in order comes the blackboard drawing, which is full size and duplicates the measurements of the miniature model proportionately.

There is naturally criticism that results in changes as these progressive steps in the design are reached. Then, following the test with the blackboard drawing, a working draft is made. This is not a permanent drawing and is made on paper, and a wood-model body is then made up like the layout of this drawing. The Murray Body Corp. calls this draft and dummy model the "mock-up".

This mock-up model is made of wood and as carriage bodies used to be made. Changes can be made by adding to or taking from the wood. Finally the design is settled, and the model is exactly as the production bodies will be later on, as far as appearance goes. The interior-trimming design, the color combination, appointments, and so on, are all settled with this mock-up and, when completed, it is the authority – the source of information – when production is undertaken. The drawing, which is changed if and when changes are made on the model, is the base for making the production drawings.

In closing this paper I have resorted to an experiment. I did not feel qualified simply to state my opinion and stand alone; I know there are debatable points that no one person can answer. Therefore, I made up a little questionnaire of twelve points, and the responses from three of the real authorities in the business are here given following each question. I selected three authorities whose interests were sufficiently divergent to cover the field as completely as possible.

*Question 1.* – What, in your opinion, constitutes the most satisfactory number of people forming the committee to pass on the new design model? Do you favor the addition of a woman member; do you think this addition an advantage?

*Answer A.* – I do not believe a committee should be so large to become unwieldy, say five or six. A woman, properly qualified, would be an advantage as women are an important factor in the purchase of new cars.

*Answer B.* – One, if he is the right person. I cannot recall a case where I thought a design was improved by adding to the number of critics. A woman's views on interiors are sometimes very helpful.

*Answer C.* – In the early stages of development very few should pass on the design proposed. Later, when the model is made in full size, finished, sales executives and manufacturing executives also should pass on the design. Until a completed stage is reached, however, the individual, unskilled in drawing effects, cannot make a satisfactory decision and, through misreading, may hamper a progressive development.

*Question 2.* – Is there a difference in results, if a wide variety of drawings more in the nature of sketches that are purely preliminary efforts and not finished pictures is employed, or is it better to show a limited number of drawings, confined to the general plan that has been previously outlined as suggestions as to the probable trend of the next design? Assume that these latter fewer drawings are made as perspective views and colored, and made as realistic as possible. Which is the best procedure?

*Answer A.* – We believe that a few, clear drawings are of more value, in this instance, than a number of sketch drawings.

*Answer B.* – I believe that, in every case, it is preferable to have well-finished and colored perspective drawings.

*Answer C.* – It is better to limit the number of sketches to a few, ranging from something similar to the existing design to something that is felt to be extreme. A few sketches, say five or six well done, should be sufficient to get a common agreement as to the trend to establish.

*Question 3.* – What is the effect or influence of last year's, or rather the previous year's, models as a controlling factor in determining the new model contemplated? Why and how, if this is an influence, does it develop?

*Answer A.* – The last year's models always influence the new design due to attempt to reduce cost of new dies and tools.

*Answer B.* – The previous year's model is usually the foundation upon which we work because of the money already invested in tools and equipment to produce it and because time seldom allows for a completely new design.

*Answer C.* – Generally, the current model exerts a vital influence on the new design since the failings or merits of the current model are well established. Change for the sake of change in itself is seldom justified, so improvement in style as compared to the current model must be shown.

*Question 4.* – Do you think that there is any benefit in having a dual control in the designing of the new model? I mean by this question, is there a benefit in having assistance that is purely artistic and not related to the industry as an influence to balance that of the conventional practical man? To what extent and in what way can such talent be best utilized in improving the results of the design?

*Answer A.* – I believe that much progress has been made in the industry by designers who are not "practical" men. This progress is due to disregard of the "possibility" to do something, and forces engineers and tool designers to deviate from set ways in search of new methods.

*Answer B.* – A capable consultant who is an automobile man and who is familiar with what is going on in the industry can be very helpful to the designing staff of an organization.

*Answer C.* – The artist should not be hampered too much in the early stages of the design. Let him think that the world is his – that the factory can build anything he wants – and that it will not cost any more than the old design did. Get some new ideas on paper where the practical man can get hold of them. It is surprising how many things can be done when they become desirable from a style or comfort standpoint.

*Question 5.* – What is the greatest influence, or rather the strongest factor that helps to decide upon the acceptance and shaping of the model design in addition to the work of the designer?

*Answer A.* – The trend of the industry, that is, the effect of more "advanced" designs, or rather of more radical designs, of other makes of cars and their effect on the public.

*Answer B.* – The whims of the management or its good judgment, as the case may be.

*Answer C.* – The factor having the greatest influence on design acceptance is its first impression as one views it. We might call this balance in design. It involves proportion, distribution, clean lines, sense of direction, and proper ornamental relief with definite individuality.

*Question 6.* – Does the period of time allowed to develop a new design limit the amount of original thought and effort applied in the making of it? What is the result of hurried planning in getting out a design?

*Answer A.* – More time undoubtedly would be desirable, but I believe there would always be this "lack of time" toward the end of the designing time.

*Answer B.* – Yes. The result of hurried planning and execution is lack of originality and rapid progress.

*Answer C.* – Time is very important in attaining a satisfactory result. It may be well to work rapidly through certain stages of the project. Then it is a good practice for those in intimate contact with it to get away from it for a short period as, many times, a new viewpoint is obtained after a relaxation from a strenuous program. Too short a time allowance will always result in many post mortems.

*Question 7.* – What value do the questionnaires and data gathered by the sales force during the preceding year have in shaping the new design?

*Answer A*:– Quite an effect as to details and minor appointments, but no great effect on the general design.

*Answer B.* – Relatively little.

*Answer C.* – Data received from customer contact and dealer suggestion should be considered carefully. It is the engineer's job to utilize those suggestions that will most benefit his clientele at large, being careful not to penalize the many for the benefit of the few.

*Question 8.* – Is the ultimate value of the design benefited by prolonged preliminary detail exactness during the successive steps between the picture drawings and the final or mock-up wood model that is exactly what the production job will be?

*Answer A.* – I believe that the ultimate design is benefited by exact duplication in the mock-up of the drawings; then the changes that are suggested can be made more easily and a truer drawing will finally result.

*Answer B.* – Yes, it pays to spend as much time and care as possible on the successive steps. It brings about a refinement of detail that is hard to get in any other way.

*Answer C.* – In a large organization an exact, or nearly so, model is an absolute necessity. First, it is necessary in order to receive the approval of a large number of executives who must pass on the design. It is more necessary, though, because the various components are widely separated in their development for manufacture and, unless there is a master model for guidance, many points in appearance will be overlooked to the detriment of the final ensemble when it reaches the production lines.

*Question 9.* – What influence in a direct way does the public acceptance of designs that are in use have upon the contemplated new design?

*Answer A.* – I believe that the effect of models already designed and their effect on the public have a great influence on new designs.

*Answer B.* – A very great influence. The management tries to decide just what makes a particular car popular. It tries to work it into its own product. Sometimes the particular feature that makes a car popular is not easy to determine; it is possible for experts to err in their decision.

*Answer C.* – The general and natural tendency of a good acceptance of current designs is to stabilize new design and to prevent radical departure. When such departure does come and it is good enough, a new style trend is established.

*Question 10.* – Do you think the preview as followed by the moving picture industry should be duplicated more extensively by body designers?

*Answer A.*—I can see no great advantage to this thought.

*Answer B.* – No. The distributors do not seem to be able to do a good job of selling the current models after they have had a look at what will be forthcoming.

*Answer C.* – Previews of new or suggested body designs cannot be held, except for those executives previously mentioned. They must, necessarily, be held far in advance of production dates and, therefore, the information must be guarded closely. The fewer people who can pass safely on a design, the less conversation will be heard about it.

*Question 11.* – Do you think that there is a lack of protection for the designer and the company interests during the time of the preliminary set-ups incidental to the development of the design?

*Answer A.* – Too many factors are involved to insure secrecy of new designs. I believe, however, that there would be a greater variety of designs among makes of cars and more original thought if greater secrecy were assured.

*Answer B.* – It has been my experience that only a few people really know just what is contemplated. Therefore, what outside sources find out from draftsmen is of no particular value.

*Answer C.* – There is never enough protection for new and advance designs. The workmen, suppliers, tool builders, and others, in a few cases, do not respect the confidence that is placed in them. In the main this condition is not true. If it were, we would have to resort to a "locking-up-the-jury" system.

*Question 12.* – Provided that the previous question states a fact, what would you suggest as a means of guarding against the practice, or rather, what protection do you think could be provided?

*Answer A.* – I do not believe that designs can be protected, but I think that individual companies would benefit if more variation among designs of different makes were developed.

*Answer B.* – Answered previously (Question 11).

*Answer C.* – To give the best protection possible, scatter out the projects, divide the job up as much as possible, limit the job as a whole to as few as possible. In this way, while one or two features may be known, no one person can put enough pieces together to make sense.

# The Construction and Operation of Six-Wheel Trucks

By Austin M. Wolf
*Consulting Engineer*

**BOTH the tractor-semi-trailer and the six-wheel vehicle have the same number of axles and wheels and each has its own particular advantages. They are seldom competitive if the transportation problem is analyzed properly and legislation does not unduly oppress either. The six-wheeler has the advantage over the tractor-semi-trailer of weight saving, more traction if four driving wheels are used, lower insurance rates, and it is free from any "jack-knifing" proclivities.**

**The chief distinction in the construction of six-wheelers depends upon the types of axles used, whether they be dead or driving. There are five classifications in use today, ranging in various combinations all the way from three driving axles to one. The rear bogie unit may have two driving axles or a driving and a trailing axle.**

**There is a natural resistance to turning in a bogie unit since the wheels do not roll tangentially when the vehicle travels around a curve. Considerable research is required on the subject of steering geometry on a six-wheeler.**

**A number of foreign trucks have applied individual wheel suspension to the bogie wheels. A foreign and a domestic design each provide an engine at each end of the vehicle, each one driving an axle of the bogie.**

**Great interest is being shown by the oil companies in the use of light-weight six-wheelers for oil and gasoline distribution because of their agility in covering greater mileage than the heavy-duty four-wheelers they are replacing.**

THE original, practical motor vehicle was built with two axles and a wheel at the extremities of each following the horse-drawn precedent. In order to carry a greater load, it was natural to consider the use of an additional axle and its two wheels. The number of wheels was increased in the case of railroad equipment to distribute the load over the rails, and it was logical that the motor vehicle should follow in these steps.

About 1911, Alexander Dow applied a second driving wheel at the end of a walking-beam. They were staggered within each driving wheel of a Mack truck, chains being used to transmit power back to the added wheels from the regular drivers. The bogie wheelbase was very short as, otherwise, steering difficulties were anticipated. This early effort proved to be premature but, subsequently, the use of three axles became popular in the form of the semi-trailer.

### Six-Wheeled Vehicles

Both the six-wheeler and the semi-trailer have their distinctive fields of operation and are seldom competitive if the transportation problem is analyzed properly. Unfortunately, there are no separate registration figures to classify completely the various types of six-wheelers or to compare their number with semi-trailers. Naturally, in States where the laws are more favorable to the semi-trailer than to the third axle, the ratio is very poor.

The ability to load or unload the semi-trailer unit without regard to the tractor gives it an advantage under certain conditions as well as in the case of special bodies of exceptional length. The maneuvering of long or awkward loads also presents an advantage and, in fleet operation, a reserve tractor provides a handier spare unit than does a powerplant which would require tying-up the vehicle while it was being installed. The "jack-knifing" proclivities of the semi-trailer are a hazard to be considered, and traction is restricted to two wheels as against a possible four in the case of the four-wheel-driven six-wheeler.

With a trailing axle it has been found that from 750 lb. to 2500 lb. can be saved in a six-wheeler of the trailing-axle type over a semi-trailer with the same body or tank, the same size tires, and all other equipment the same. This saving in weight is due to the elimination of the upper and lower fifth wheel and the king-pin, the overlapping of the trailer and

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tractor frames, the landing gear, and the reduction in overall length of 4 to 5 ft. One firm using a trailing-axle equipment found that, since New England laws allow 40,000 lb. gross on either six-wheelers or tractor-semi-trailers, they are able to haul about a ton more payload on these units than would be possible on a tractor-semi-trailer. The added 2000 lb. amounts to an extra $12 per trip assuming a hauling rate of 60 cents a hundred and, if we figure conservatively that these trucks will make 250 trips a year, this added load would mean an additional income of $3000 per year per truck hauling only the same 40,000 lb. gross that the trailers would haul. In addition, since a truck travels a considerable part of its life unloaded, this same 2000 lb. is not hauled by the six-wheelers during their unloaded life, resulting in a further saving.

The semi-trailer has often been sold on the fallacy that one can pull more than one can carry. Attachment axles have sometimes been fostered as a means of endowing a light truck with the abilities of a large one. In any vehicle the vital factors are engine output and gross vehicle weight. Efficiency enters into the picture in but a small way assuming equal design and workmanship in the different types of construction. Although there are a greater number of parts in the case of a four-wheel-drive bogie unit than in the case of the four-wheel-drive truck, it must be remembered that, in the former, the work is divided among a greater number of smaller operating parts. The use of multiple units in various engineering fields has indicated successful operation, often better than in the case of a single unit. The additional friction that might exist is exceedingly small.

Insurance rates are distinctly favorable to the six-wheeler as compared with the tractor-semi-trailer. Liability and property damage insurance is based on a multitude of variables. Let us assume that a six-wheel truck will carry a five-ton load; that it will be privately owned and used in private work; and that it will not move regularly more than 75 miles from its regular station. With these limitations in mind, coverage is as follows:

| | | |
|---|---|---|
| For New York City, Liability | $270 | Property Damage $104 |
| For Yonkers, N. Y., Liability | $116 | Property Damage $ 57 |
| For Paterson, N. J., Liability | $140 | Property Damage $ 79 |
| For New Brunswick, N. J., Liability | $ 91 | Property Damage $ 52 |

To these rates must be added 25 per cent in each instance for a semi-trailer which has a body not over 25 ft. long and which will not carry more than 7 tons. If the carrying capacity of the trailer is in excess of 7 tons, 35 per cent must be added to the six-wheel rate noted previously. If the semi-trailer is more than 25 ft. long, 100 per cent must be added. Furthermore, it is to be noted that in like territory there is no difference in rates between the four- and six-wheeler of equal tonnage.

The distinction between the semi-trailer and the six-wheel vehicle is best defined by the English custom of referring to the former as an articulated six-wheeler and to the latter as a rigid six-wheeler. A six-wheeler might be defined as a wheeled vehicle incorporating a single, rigid frame-structure for powerplant, driving mechanism, cab, and body carried by three axles, at least one of which drives and, in which, the weight distribution on two adjacent axles is substantially constant. Truck developments in Europe already have incorporated individual wheel suspension, thus possibly requiring a more comprehensive definition stating that the vehicle is "carried by six-wheel spindles in normally paired transverse alignment, and so on." The number of individual wheels is omitted purposely to avoid confusion between single or double wheels or single or dual tires. The earliest six-wheelers were equipped with solids, then single pneumatic tires, but the greater carrying capacity of duals made them immediately popular.

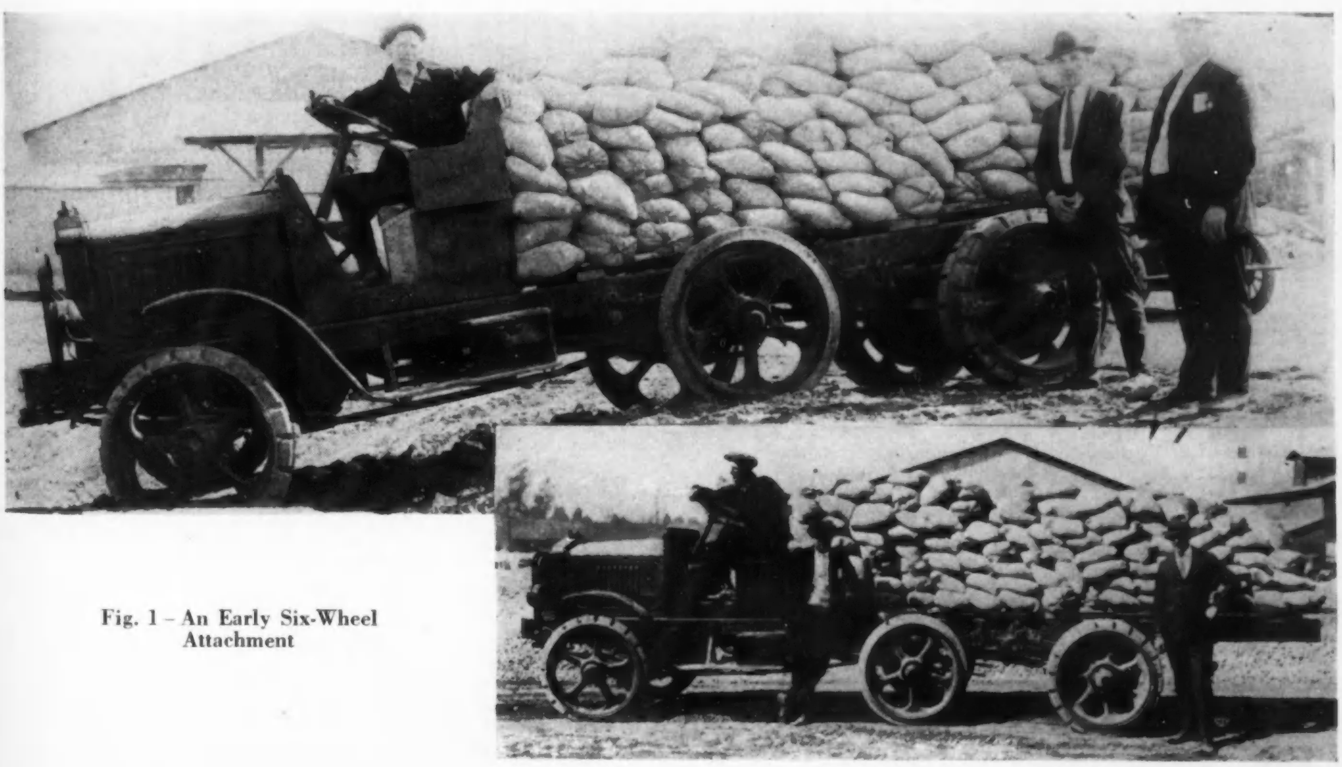


**Fig. 1 – An Early Six-Wheel Attachment**

In the transportation field covering trucks and buses, the Pacific Coast has been the birthplace of many notable developments, and it is here that the six-wheel-truck attachment was inaugurated. The semi-trailer was enjoying popularity, and the California motor-vehicle laws had been changed to permit a gross load of 34,000 lb. on three axles, while the two-axle truck was restricted to 22,000 lb. This restriction proved to be the incentive for the installation of a third axle under a conventional chassis and, in the early part of 1923, David L. Van Leuvan in Los Angeles conceived the application shown in Fig. 1 on a standard truck in which a steering axle was placed a considerable distance ahead of the rear axle. This equipment was produced by Six Wheels, Inc. Intermediate and rear axles had their own semi-elliptic springs which were interconnected at their adjacent ends by an equalizing bar.

To eliminate the complication of the added steering control it was realized that, if the wheels were placed closer together, it could be eliminated. The Model "C" Maxi six-wheel unit was developed in 1924 in which a trailing axle was placed closely behind the driving axle and connected to the rear end of the regular rear springs by means of a frame-supported laminated-spring equalizer. A leaf-spring radius rod also was used at each side, the width of the plates being vertical. The "KA" unit followed, consisting of a rigid, frame-supported equalizer supporting the extra wheel on an integral cantilever spindle. This construction, however, did not comply with legal requirements as to controlled-weight distribution, and brakes could not be applied properly due to absence of torque control. Fager and others continued the attachment development on the West Coast. Intensive work on the four-wheel-drive bogie began with efforts of the Goodyear Tire Co. around 1918-1919.

### Legislation

In 1932 a number of States recognized the virtues of the six-wheeler and permitted increased capacity. At that time the permissible maximum gross loads varied between no restriction, 48,000 lb., and 16,000 lb. Fifteen States gave no preference over the four-wheeler and, where preference did exist, the allowance varied.

On Oct. 15, 1935, legislation allowed gross weights as depicted in Fig. 2. It will be seen that the situation has improved considerably, although much still remains to be done in many of the States to give the six-wheeler its due recogni-


Fig. 2 – Legislation Covering Six-Wheel Vehicles

**Table 1 – Axle Types, Six-Wheelers**

| Group | Front Axle | Center Axle | Rear Axle | Designation |
|---|---|---|---|---|
| I | Dead | Driving | Dead | $6 \times 2_c$ |
| II | Dead | Driving | Driving | $6 \times 4_{c+r}$ |
| III | Driving | Driving | Dead | $6 \times 4_{f+c}$ |
| IV | Driving | Driving | Driving | $6 \times 6$ |
| V | Dead | Dead | Driving | $6 \times 2_r$ |

tion based on the lower road impact-value tests run by the office of Public Roads, Department of Agriculture. Other than the most logical method of computing gross loads by the formula of the American Association of State Highway Officials, $W = c(L + 40)$, and which ten States utilize in their legislation, the top limit is now 40,000 lb. in seven States, with decreasing increments in other States until we reach 20,000, 18,000 and 16,000 lb. in two States each and 7000 lb. in Louisiana and Texas for any vehicle. Permissible axle and tire loadings dictate the restrictions in a number of States. No preference is given the six-wheeler over the four-wheeler in Alabama, Florida, Kentucky, Louisiana, Mississippi, Oklahoma, South Carolina, Tennessee, and Texas.

There seems to be a need for a definition of a six-wheeled vehicle in many State laws in so far as weight distribution on the rear axles is concerned. Many of us have noticed the hybrid six-wheelers that are running over New Jersey highways in which a third axle is attached ornamentally to the rear end of the truck frame; this axle might as well be hung on the cab roof since there is only one theoretical point and then under static conditions only that each of the rear axles would take their proper proportion of the load. Practically this point never exists.

### Present Constructions

The chief distinction in six-wheelers depends upon the type of axles used, whether they be dead or driving. Table 1 indicates the various types and follows the chronological order of commercial developments. *Group I,* with the center driving axle, has been enjoying the greatest popularity in both in-built and "attached" rear axles. Simplicity of construction, reduced cost and weight, and lower maintenance make it desirable where special tractive ability is not essential. *Group II,* with two rear driving axles, was developed for high tractive ability, but with the inherent disadvantage, as compared with *Group I,* of greater expense, complication, and weight. *Groups III* and *IV* are evolutions of the four-wheel-drive truck combining its virtues with the rear bogie developments. *Group V,* with the driving axle at the rear, was sponsored by Federal. Since most bogie designs provide a swiveling or equalizing action, the load is raised only one-half the height as in the case of a four-wheeler, assuming that the equalizer is centrally pivoted. The lessened shock, particularly at increased speed, is very noticeable.

Under the heading "Designation" in Table 1, the first number refers to the total number of wheels on the vehicle and the second number gives the number of driving wheels and their location, the sub-letter indicating whether the drivers are on the front, center, or rear axles or any combination thereof. A four-wheeler can be similarly designated, $4 \times 2_r$. A front drive would be $4 \times 2_f$, and a four-wheel drive, $4 \times 4$.

### Weight Distribution

When the driving and dead axles are combined in a bogie unit, greater weight is usually borne on the driving axle to

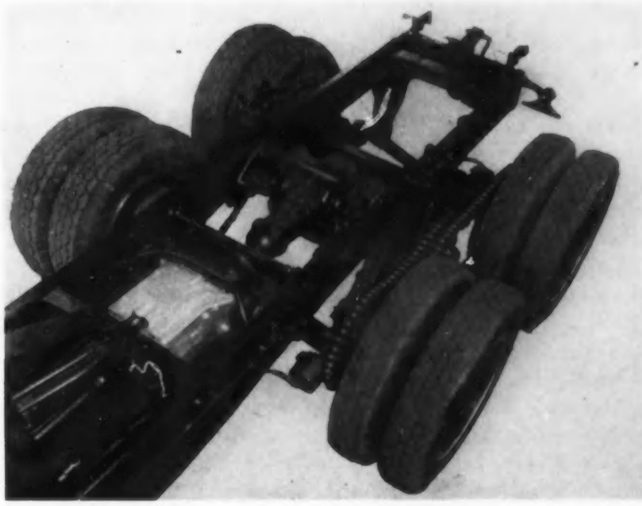


Fig. 3 – Sterling Dual-Chain Drive

increase its tractive ability. When two driving axles are used, the rear-end static weight is divided equally between them. In such a case the desirable weight distribution on the front, center, and rear axles is 20 per cent, 40 per cent, and 40 per cent. In this way the tires are loaded equally, this statement being based on dual tires on the center and rear axles. When the six-wheeler was first developed, solid tires were prevalent and, by the proper sizing, it was possible to throw more than twice the front-axle weight on the center and rear axles. The weight on the driving tires for traction varies of course with the application of propelling torque altering the static-load distribution. Generally speaking, this static-load distribution for traction assumes the following proportions:

| | Per Cent |
|---|---|
| Camel-back, four-wheeled truck | 67 |
| Conventional-type, modern four-wheeled truck | 70 |
| Older-type, four-wheeled truck | 75 |
| Conventional, tractor-semi-trailer | 42 |
| Conventional, two-wheel-drive six-wheeler | 60 |
| Camel-back, two-wheel-drive six-wheeler | 44 |
| Conventional-type, four-wheel-drive six-wheeler | 80 |
| Camel-back, four-wheel-drive six-wheeler | 80 |

### Various Constructions and Drives

Following the introduction of the Mack double-jackshaft design, Sterling Motors Corp. introduced a dual-chain-drive chassis ($6 \times 4_{c+r}$) shown in Fig. 3 in which a single jackshaft drives all four wheels by means of a double sprocket at each side, the inner one driving the forward wheels and the outer one driving the rear wheels. The three-in-one differential used is of the Krohn type and has a middle differential distributing the power to each side. A similar power divider at each side permits equalizing the drive between the two wheels on that side by controlling a co-axial tubular and solid jackshaft on the end of which the sprockets are located. The axles are free to conform to any road contour since the spring ends merely rest on the axles and pivot about the center of the main housing. Four longitudinal radius rods hold the wheels in their proper position fore and aft, while a transverse radius rod with ball-and-socket joints locates each axle crosswise.

The Timken walking-beam construction with each end spherically capped around the axle housings is used on the

heavier designs but, in the lighter models, a parallelogram effect is obtained by four rubber-bushed tie-rods at each side ($6 \times 4_{c+r}$). While a through worm drive is used on the center axle and the regular worm on the rear axle, Wisconsin double-reduction carriers can be substituted, the forward one having a through bevel pinion shaft. In the lightest models, the upper tie-rods are replaced by a single, centrally located member. This bogie also can be obtained with a trailing tubular-type axle ($6 \times 2_c$).

*Group II* in Table 1 can be subdivided further into a classification in which a through drive or an inter-axle differential is used. The worm drive was quite popular in 1932, but it is fairly extinct today in favor of the double-reduction drive. Where wheel fight has occurred in the case of no inter-axle differential, worm wheels have been destroyed from overheating whereas, in the case of the double-reduction drive, the sliding action of the gears is limited so that high-temperature effects would be kept to a reasonable minimum.

In 1932, Hendrickson Motor Truck Co. was building two types of construction each employing the characteristic Hendrickson walking-beam with the ends suspended below the axle housings. In one design, a transfer case was provided with a third differential to distribute the drive to the rear axles. The International Harvester Co. is licensed under the Hendrickson patents, and it will be noted at the right in Fig. 4 ($6 \times 4_{c+r}$) that a practically similar layout to the Hendrickson is used, the chief difference being that, whereas Hendrickson previously supported the front end of the rear propeller shaft on the center-axle housing, International is using the later Hendrickson design in which it is supported in a swivel support on the frame. The use of four joints enables the angle to be divided among them and keeps the individual angularity down to a minimum. The pivoted support is mounted over the center-axle housing and does not touch it unless it goes up about 6 in. A coil spring is used to prevent rattling, and the weight of the shafts normally keeps the support in its lowest position. In the upper view of Fig. 5, the low-disposed walking-beam trunnion is discernible. The lower view shows the transmission and power

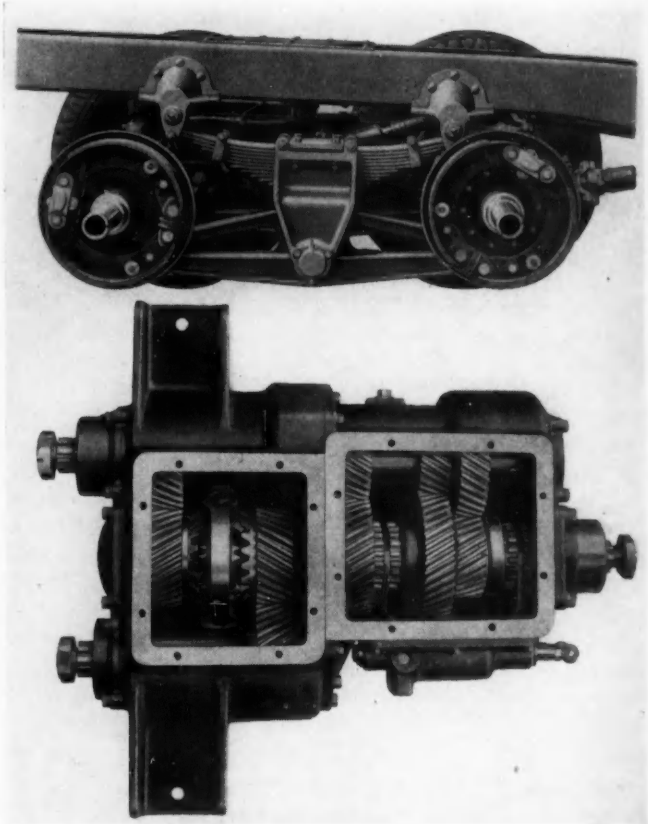


Fig. 5 – International Walking-Beam (*Upper View*) and Power-Dividing Transmission (*Lower View*)

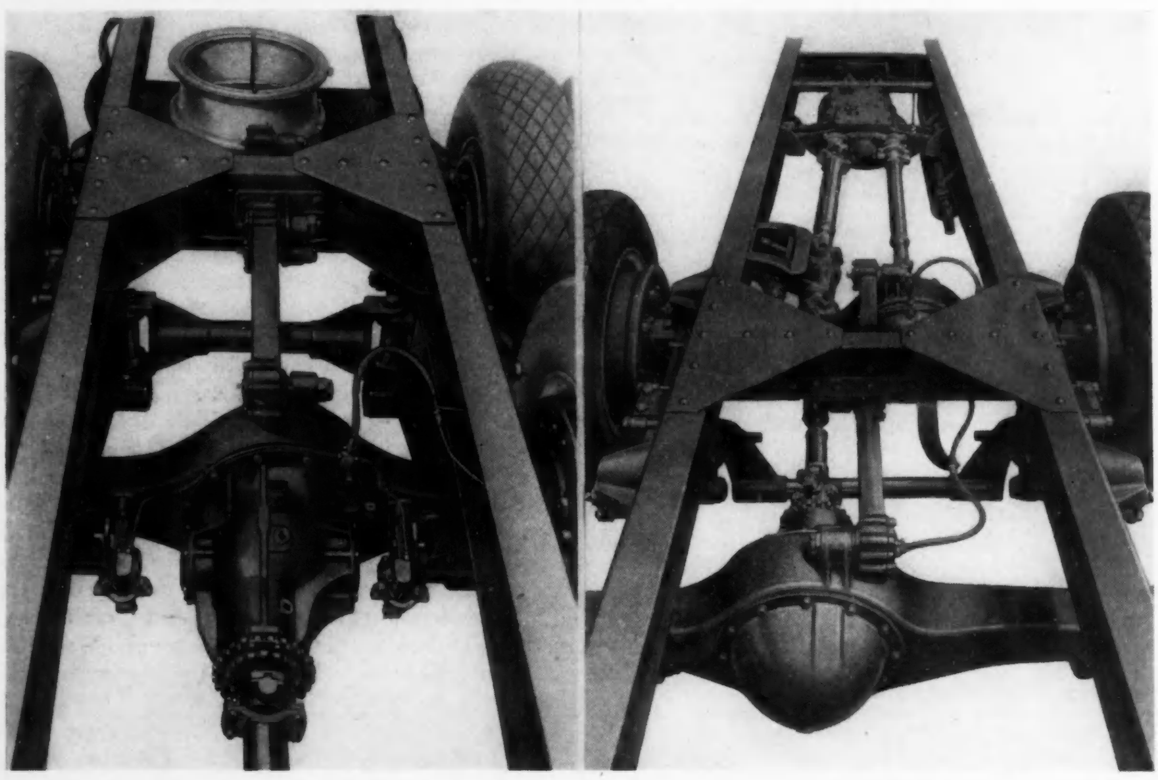


Fig. 4 – International Six-Wheelers; (*Left*) $6 \times 2_c$; (*Right*) $6 \times 4_{c+r}$

divider bolted together to form one unit. A differential lock can be furnished on request. International also can provide a tubular rear trailing axle (see left view, Fig. 4) in which case the center-driving-axle differential housing is located centrally ($6 \times 2_c$).

The through-shaft drive type of construction ($6 \times 4_{c+r}$) was introduced by Mack in 1932 in which the power divider is mounted on the forward-axle housing shown in Fig. 6. Torque arms extend from each housing to a special cross-member. Individual longitudinal radius rods position the axle ends fore and aft, and a transverse radius rod locates each axle transversely. The springs are centrally trunnioned, and their ends are provided with hemispherical bearings in corresponding sockets which swing freely on the axle banjo.

The greatest problem in six-wheelers has been the maintained positioning of the bogie unit in alignment with the frame. The sole use of a central trunnion results in a tremendous leverage on it due particularly to the transverse forces imposed at the tire contacts. The Mack and Sterling transverse radius rods have aided materially along this line, and there is no doubt that a wishbone type of radius rod may be developed to cope with forces in the two directions.

The Mack construction shown in Fig. 6 provides a through shaft extending back from the power divider and which drives the rear axle through a relatively long propeller shaft. The relationship of the various parts of the bogie is shown clearly in Fig. 7.

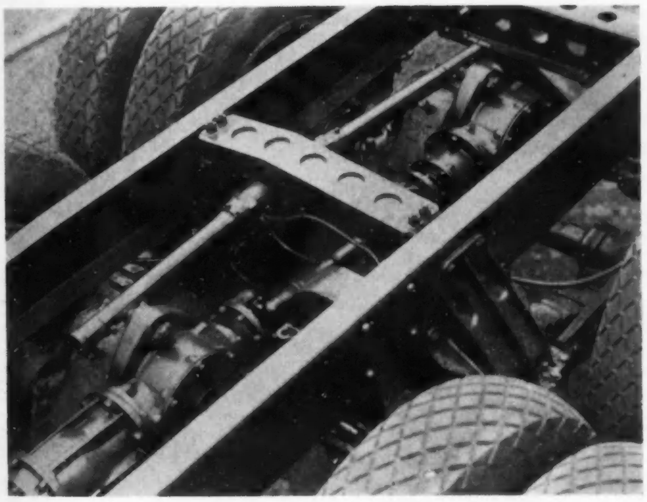


**Fig. 6 – Mack Six-Wheeler with Power Divider on Center Axle**

Federal Motor Truck Co. produced two types of six-wheel designs with an all-drive bogie ($6 \times 4_{c+r}$) and the single-drive-axle type ($6 \times 2_r$). Referring to the former type, power was conveyed to a rear driving axle by means of a propeller shaft connected at its forward end to a pinion driven from the center-axle ring-gear and connected at the rear end to a pinion driving the rear-axle ring-gear on the opposite side to compensate for the reversed rotation imparted to this propeller shaft. The single-axle-drive bogie had a central trailing axle to take advantage of the added loading on the rear axle due to torque reaction. Springs were eccentrated to the rear to increase the static loading. The parallelogram in the first Federal designs was provided by tandem springs and without the use of radius rods. In the later designs, double bushed radius rods replaced the lower spring, and the upper spring ends were shackled universally. The combination of an upper trunnioned spring and separate radius rods previously was used by Dodge ($6 \times 2_c$).

### Four-Wheel Truck Attachments

While some of the early attachments did not cope with all the problems involved in a six-wheeler, their status today is such that, when properly designed and engineered into a particular four-wheel truck design, they are able to give entirely satisfactory performance and are being used to a considerable extent. The original Maxi unit has evolved into its present form as shown in Fig. 8 ($6 \times 2_c$). The Truxmore unit, shown in Fig. 9 ($6 \times 2_c$), consists of a walking-beam on each side with "gravity spring suspension" connected directly to the rear or trailing axle and to the rear end of the suspension spring to which it is adjustably connected by a worm and nut construction so as to give one of three locations resulting in a 50/50 ratio, 55/45 or 65/35, the first figure referring to the driving-axle load distribution. A laminated-spring type of radius rod with the width of the leaves vertical is used to absorb brake torque. A laminated-spring type cross-member replaces the pivot anchorages of rigid cross-members in other constructions.

The Trucktor unit in Figs. 10 and 11 ($6 \times 2_c$) uses a short equalizer between the downwardly extending shackles from the regular suspension spring and the trailing-axle spring. A V-shaped yoke acts as a towing member and the brake-torque unit, being rubber bushed on a tubular cross-member. The spring seats are bushed on the axle and, by using a slipper type of end at the rear of each trailer spring, a certain amount of tracking motion is possible. Practically any weight distribution up to 50-50 per cent can be furnished, although it has been found by this company that the best operation conditions are secured with a load distribution of 57-43 per cent.

The Grico unit ($6 \times 4_{c+r}$), produced by the Gear Grinding Machine Co. and shown in Fig. 12, consists of two driving axles mounted on tandem springs. A two-speed transfer case is mounted in the frame and receives the short torque tubes of each axle within whose spherical seats are located in each instance a Rzeppa constant-velocity universal joint. An inter-axle differential can be furnished. The Thornton unit is of similar design.

### Brake and Propulsion Torque Reaction

In the various designs that have been enumerated, the transmission of brake and driving torque is dependent upon either torque rods, springs, or a combination thereof. The matter

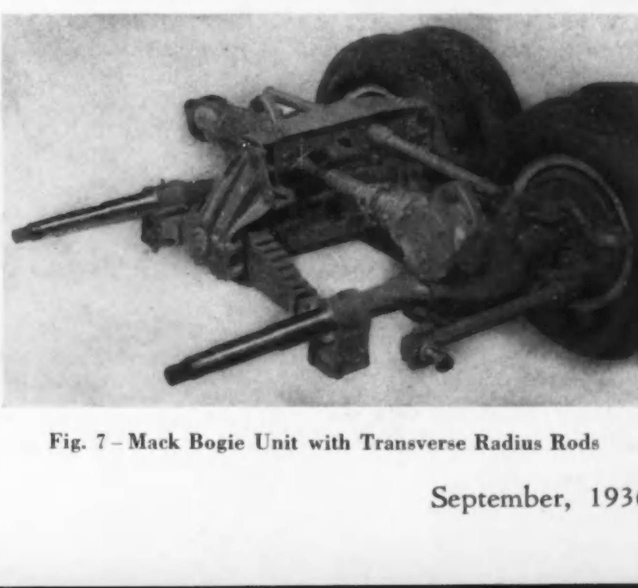


**Fig. 7 – Mack Bogie Unit with Transverse Radius Rods**

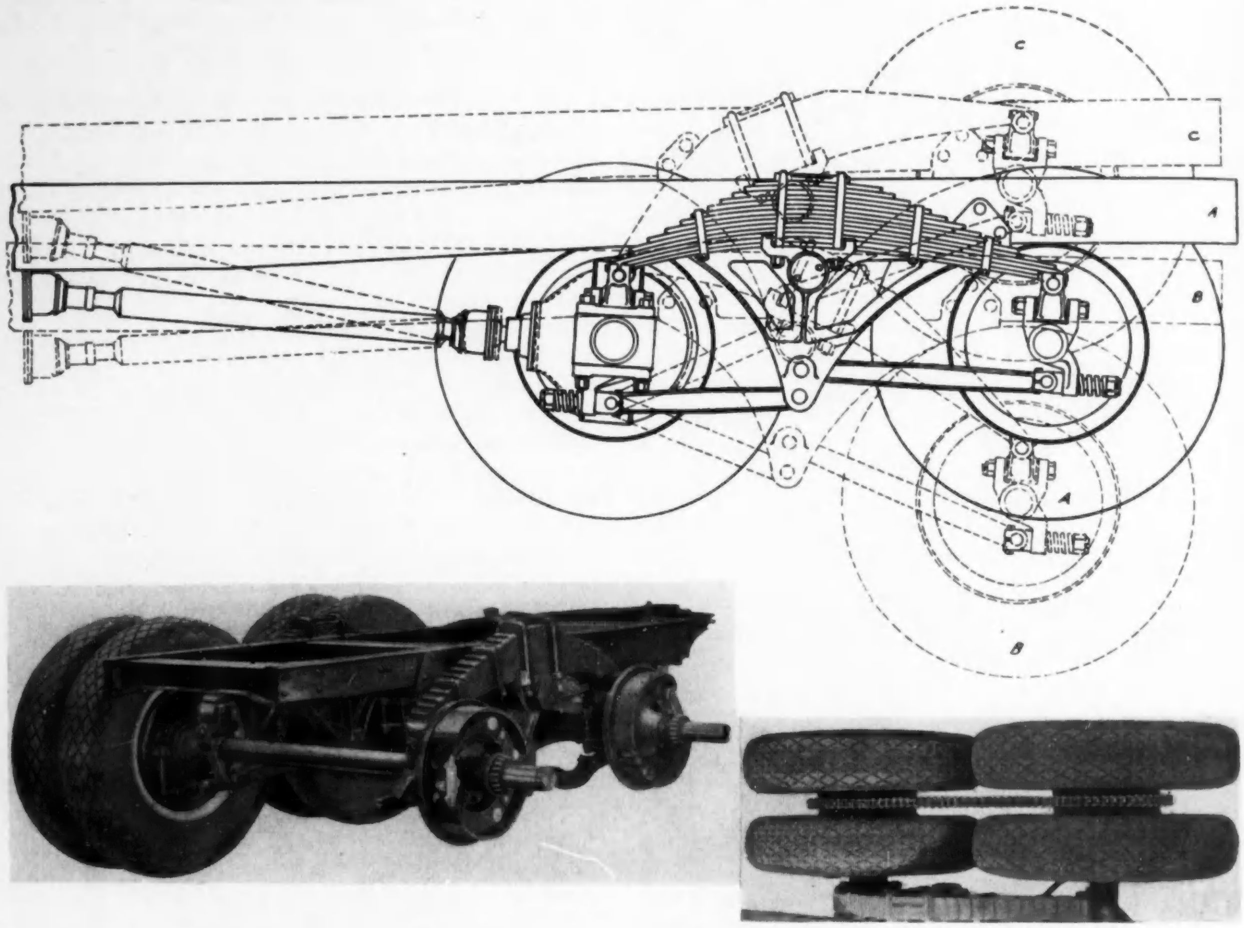



Fig. 8 – Maxi Unit Showing Flexibility of Movement and Chain Attachment

of torque reaction is of great importance and has considerable influence on maintenance costs, particularly those of tires. In braking, the wheels of the center axle tend to dig in, whereas the driving-torque reaction depresses the wheels of the rear axle. Since the trunnion type of suspension is almost universal, a study of the forces involved will be of interest.

Referring to the upper view of Fig. 13, a relatively high trunnion near the top of the tires is indicated. The force $G$ represents the vehicle weight carried on the trunnion. When the brakes are applied, the body tends to slide forward off the bogie, creating a horizontal force $H$ on the trunnion in the direction of vehicle motion. The force $H$ produces an overturning moment on the bogie, which moment greatly increases the normal upward force $N_1$ under the center tires and accordingly decreases $N_2$ at the rear. The resultant of $H$ and $G$ "pierces" the surface of the road (a line going through the contact points of the tires) at the point $P$. The fact that $P$ is much nearer the center tires than the rear ones means that the load $G$ can be considered as being shifted forward and downward to $P$ at a distance $C_1$ from the center tires instead of at the distance $A_1$. This diagram makes it evident that $N_1$ must be greater than $N_2$.

The resultant increased deflection of the center tires reduces their effective rolling radii and, accordingly, increases their r.p.m. The effect is reversed on the rear tires where the load is reduced, the deflection increasing their rolling radii and the tires tend to rotate at a slower speed. If there is a direct-drive connection between the center and rear axles, this trend toward a difference in r.p.m. of the center and rear tires cannot exist without "breathing" of the various resilient elements in the bogie unit including the shafts and/or slippage. If slippage does occur, tire wear ensues.

In the lower view of Fig. 13, the trunnion is located at a lesser distance $R$ from the road and the point $P$ is moved back. Analytically, but of course not practically, it would be desirable to reduce the distance $R$ to a very small amount. The virtue of a low trunnion center is here proved.

The application of driving torque has a similar effect but in the opposite direction, and it will thus be seen that it would be impossible to eccentrate the trunnion in favor of one condition without aggravating matters in the other. Furthermore, the torques are continually variable.

### Inter-Axle Differentials

There probably has been more argument on the subject of inter-axle differentials than on any other phase of the six-wheeler when two driving axles are used in the bogie unit. Theoretically at least, as explained in connection with Fig. 13, the use of such a differential is vital. Variations in the rolling radii of the tires will, in the absence of a differential, cause one axle to "fight" the other, resulting in mechanical and tire wear. Yet there are many who claim that satisfactory results can be obtained without the differential if the operator will watch carefully the inflation and rolling radii of the tires,

and for this reason they feel that the added complication of a differential is not warranted.

Should a center wheel, for instance, pass over an obstruction while the rear wheel is still traveling on the level, a greater distance is traversed by the former wheel, and we have the same argument and condition that occur in a four-wheel drive truck where, without an inter-axle differential, a "breathing" action in the torque hook-up, flexure in the driving mechanism, and possible slippage of the tires must occur. There can be no argument against the fact that, without the differential, back-lash is not taken up until creep accomplishes it, during which time but one axle does all the driving.

The inter-axle differential overcomes these difficulties but at the expense of weight, complication, and initial and maintenance costs. This statement, however, is based on the assumption that an ideal differential is available. Slippage of one wheel renders the entire bogie unit inoperative. In a through drive stalling is impossible if two wheels on one axle have sufficient traction. For such reasons special differentials or substitutes have been developed such as the Timken "high-traction" type and the Krohn compensator. Some of these units have cast doubt on the ability of the ordinary differential to function satisfactorily at the speed at which it must operate (propeller-shaft speed) since a time factor enters into the action of such a differential.

Should the inter-axle differential place the full driving torque on one axle, the mechanism must be as robust as that of a single axle to take care of it. An equal power distributing differential permits each axle to take its half of the load, reducing the size of the parts and their weight. The Patent Office records are replete with attempts to tame the uncontrolled differential, and there is unquestionably a demand for such results extending even into the passenger-car field.

One solution of this problem has been the Mack design of a power-divider, the terminology indicating a justified contempt for the ordinary term. The mechanism is shown in Fig. 14. The transmission drive is conveyed to a cage carrying two sets of radially held plungers. They contact on the inside with a double cam, the high points of which are staggered and the use of two rows of plungers results in a positive drive. The inside cam is connected to the rear-axle drive, and an outside corresponding cam surrounds the round-ended plungers and is splined to the spiral pinion of the center axle. The shape of the cams can be varied so that equalization can be made similar to that of a regular differential, or they can be designed for the other extreme of complete locking. In the Mack design the equalizing action favors the axle having good traction, and yet stops short of a degree of locking that would overstress any of the driving parts.

### Brakes

The early six-wheel movement suffered considerably from inadequate or improperly matched brakes as used on the trailing axle; this difficulty also happened in the semi-trailer field. Considerable improvement has been incorporated in recent designs in attempting to give the proper braking-effort distribution between the center and rear axles. "Digging in" of the bogie is considerably greater in braking than in the application of driving torque due to its greater severity. For this reason, identical brake equipment on all four wheels is not desirable. Mack designers early appreciated this condition and coped with it by using a larger Westinghouse air-brake chamber to actuate the center-axle camshafts and a smaller one at the rear where less traction is available during retardation.

Most factory-built six-wheelers with hydraulic brakes generally provide a cylinder for the rear axle approximately two-thirds the area of the operating cylinder on the center axle

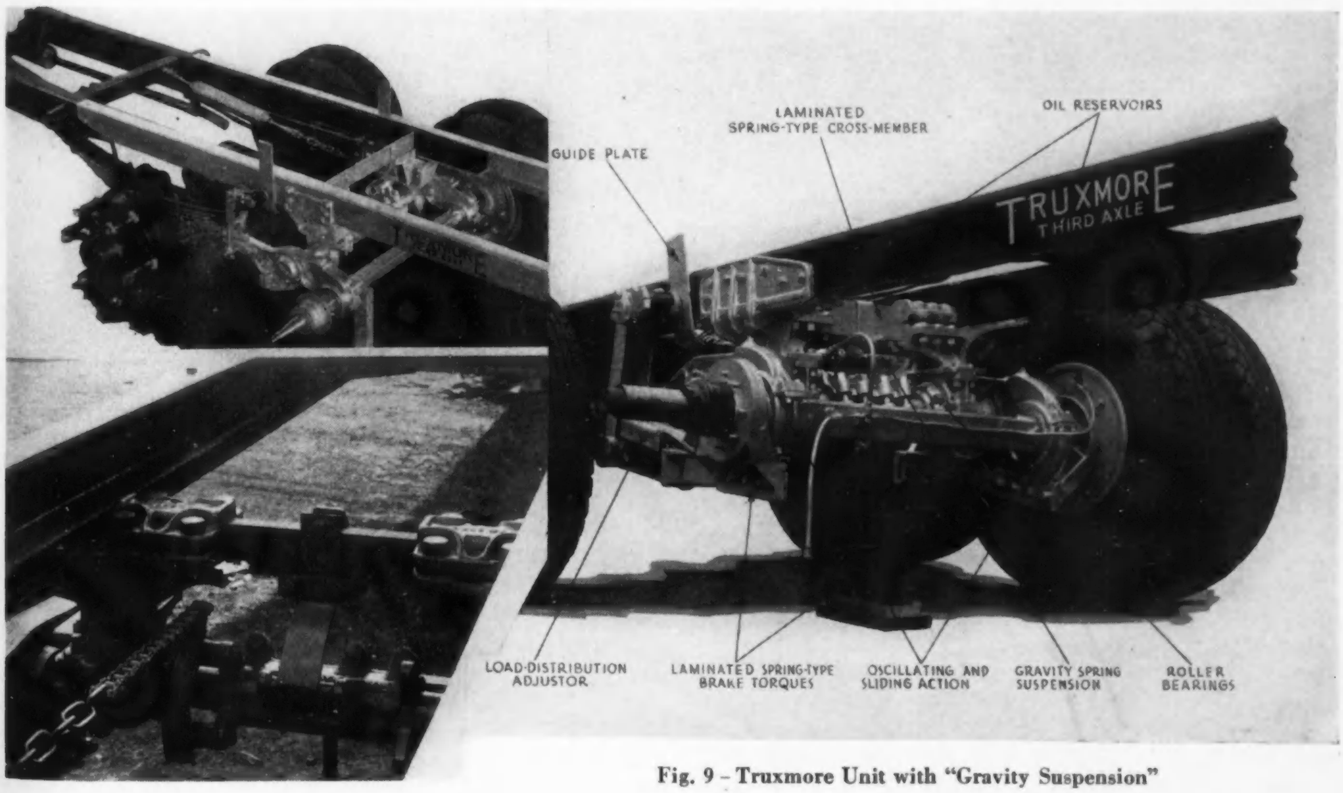



**Fig. 9 – Truxmore Unit with "Gravity Suspension"**

to prevent the otherwise premature locking of the rear wheels. When the third-axle manufacturer supplies two rear driving axles to the operator, it is customary to use a smaller brake on the rear axle and also a smaller operating cylinder. For instance, should the center axle be equipped with a 16 in. x 3 in. brake, the trailing axle will probably carry a 16 in. x 2½ in brake. When a trailing axle is adapted to a four-wheel vehicle by a third-axle manufacturer and must be installed by distributors who are more or less familiar with the whole problem, results will vary greatly. The main difficulty is due to the fact that the operator will generally purchase the smallest truck available on which the braking equipment has no great margin of over-capacity for the extreme loads he intends to carry and then, with the addition of a third axle, he expects to carry 20,000 to 25,000 lb. on the vehicle.

When hydraulic brakes are applied to this type of truck, a barrel type of master cylinder is probably mounted in the motor arm and on which there is no chance of changing leverages or cylinder sizes without considerable engineering work. Rather than permit itself to be a party to field installations from which only mediocre performance can be expected, the Hydraulic Brake Co. is suggesting that the trailing axle be treated in the same manner as in the case of the semi-trailer. The trailing axle is provided with a master cylinder and a small booster of its own which is actuated by a valve mounted in the foot-pedal-rod line of the four-wheel braking system. This arrangement does not disturb the manufacturer's original brake equipment and there can be no divided responsibility for the performance of the complete installation. Even this type of installation will vary greatly in its effectiveness since some truck manufacturers still cling to stamped drums. Without a third differential the rigid hook-up makes it possible to obtain relatively good braking without any careful balance of braking effort between the brakes of the two axles. This advantage is secured, however, at the cost of severe strains upon the axle shafts, the gears, and bearings. In other words, an imperfect degree of equalization will give results that could not be tolerated in a bogie with an inter-axle differential.

### Tires, Steering Response, and Traction

From the inception of the six-wheeler it has been noticed that the rear tires have suffered unduly by debris loosened by the center tires and thrown immediately under the rear ones, resulting in cuts and punctures. Railroad spikes have been picked up and found inside the tubes. Such experiences have caused some operators to use puncture-proof tubes. The four-wheeler encounters no such trouble. No one realizes what a road holds until a search is made with an electromagnet which, of course, does not indicate all the non-ferrous material which is of no interest to the magnet. Tires are also liable to suffer from unequalized brakes due to the greater number of brake units. For this reason the more sensitive braking problem cannot be ignored.

A comparison between the two-wheel-drive six-wheeler (6 x 2) and the four-wheel-driven type (6 x 4) discloses the fact that, unless the driving tires are to be overloaded materially, it is necessary to reduce the load that can be carried on a given set of tires. This necessity is due to the fact that a two-wheel-drive six-wheeler (6 x 2) almost always has a greater proportion of the load upon the driving wheels. In

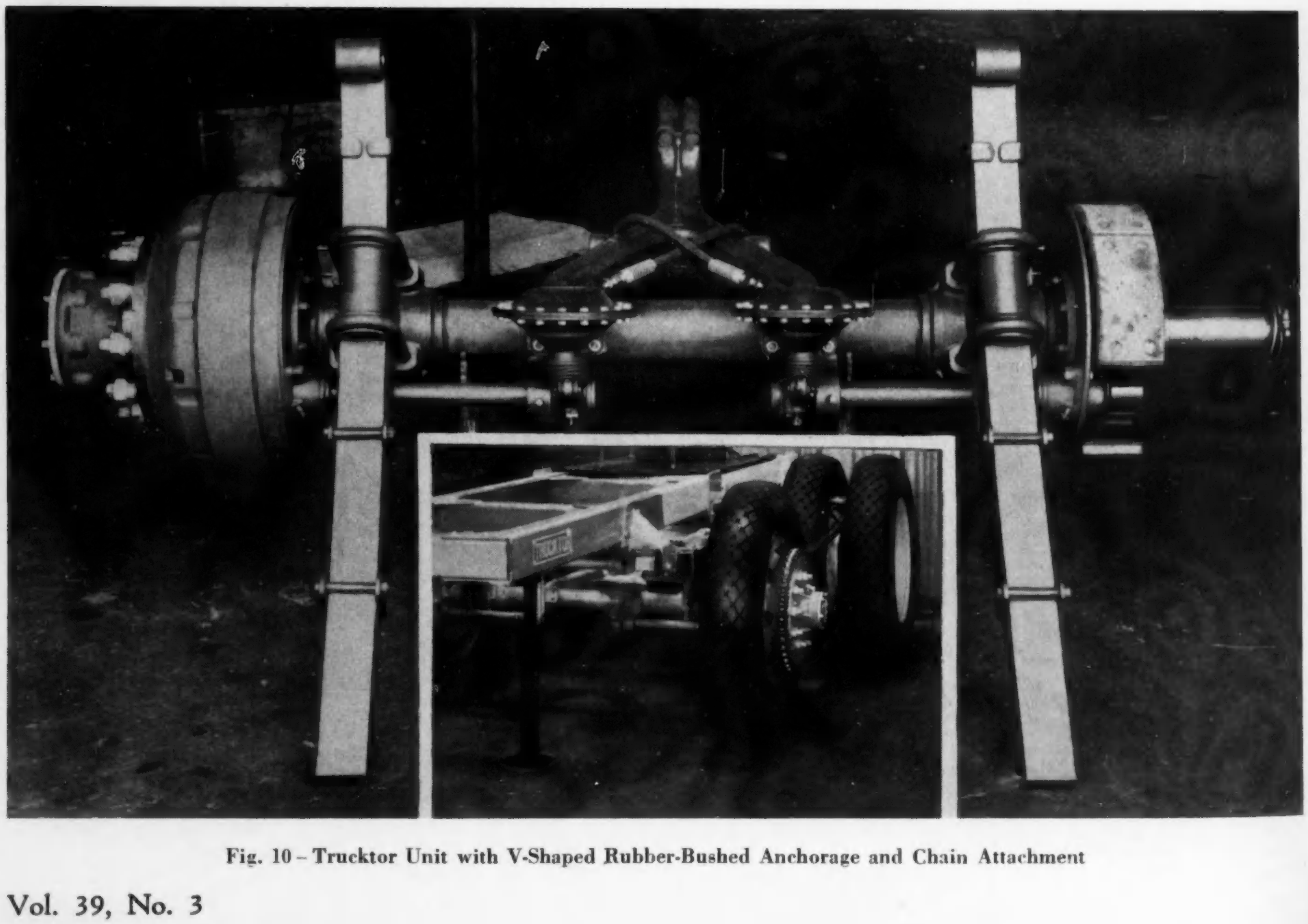


Fig. 10 – Trucktor Unit with V-Shaped Rubber-Bushed Anchorage and Chain Attachment

other words, the practical gross vehicle weight that a four-wheel-drive six-wheeler (6 x 4) may carry is about nine times the capacity of one tire whereas, in the two-wheel-drive six-wheeler (6 x 2), the weight would be approximately eight times the capacity of one tire, which relation also holds for the tractor-semi-trailer. The difference in the case of the former is occasioned by the underloadings of the trailing-axle tires whereas, in the latter, it is due to the under-loading of the front tires.

We are all cognizant of the theoretical steering geometry of the conventional four-wheeler. Knowing that a vehicle bounces over the road and that the tires are not "geared" to it, it is realized that the theoretical ideal is not maintained and we have been satisfied in the case of the six-wheeler to assume that the turning center is on the prolongation of a line passing approximately through the trunnion axis. Furthermore, we know that a vehicle or body tends to gyrate about its center of gravity which does not lie on this line. The center wheels will tend to roll more truly than the rear. Whether there are two or four driving wheels should influence results.

It has been shown[1] that there is a greater difference in the distance covered by adjacent dual tires on an axle end than in the case of one tire ahead of the other on a bogie unit, assuming that each vehicle swings around the same radius. This conclusion, however, does not take into account the side-scuffing tendencies which exist and which fortunately are mitigated to some extent by the tread design. Side-scuffing must occur when a tire is rolled through an arc since the momentary contact area with the road is shifting continually sideways as a new area presents itself to the surface of the road. In the six-wheeler, conditions are worse since none of the tires have a true tangential roll. Even if one set were allowed to do so, the other set would suffer all the more. Therefore there is a natural resistance in a rigidly hooked-up bogie to a change of direction. The entire subject of steering geometry and control would bear profitably considerable research work. There is no doubt that some complaints in the past of hard steering are traceable to this cause. Conversely the recent severe winter with ice-bound roads very often gave indications of lack of steering response, and a critical condition arose when the front-axle loading became less than 16 per cent of the gross vehicle load unless chains were used on the front wheels.

Experience has shown that it is desirable to have not less than 14 per cent of the gross load on the front axle on wet pavements. These figures are based on vehicles showing the least steering response. Chassis details have a decided influence in this regard. In one particular case it was found that a smaller percentage on the front axle was permissible when the fixed eye of the front spring was located at the rear and the shackle at the front. Hard steering is no doubt aggravated in some four-wheel-drive bogie designs without an inter-axle differential.

The front-axle weight minimums were obtained by the use of tank trucks which permitted considerable experimental variation in the emptying or filling of the various compartments.

Poor steering response also can result in overloading the front axle, but we are familiar with this phase in connection with the four-wheeler. The problem of poor response is exaggerated in the case of the camel-back design of six-wheeler because the weight on the front axle is usually greater and,

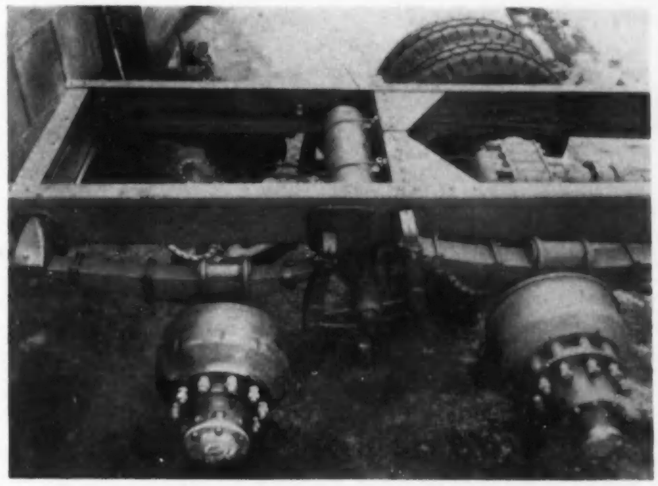


**Fig. 11 – Trucktor Equalizer Bar and Suspension Layout**

since its distance to the bogie is short, the front wheels have a smaller lever arm through which to overcome the resistance of the bogie to turning.

Although some operators have found that a barefooted six-wheeler with a trailing axle (6 x $2_c$) has given better results than a four-wheeler (4 x 2) with chains over snow, others have complained of lessened traction, especially over muddy roads. Again chassis details must be considered as the third-axle hook-up has a great influence in such cases. The recent severe winter has brought out the advantages of driving on all four bogie wheels. It was common practice at the peak of the storm period to have two-wheel drives (6 x 2) lie in wait at roadside stands until a four-wheel-driven six-wheeler (6 x 4) came along to break the road. Operators have stated that a four-wheel-driven six-wheeler will plow through under conditions where a four-wheel truck cannot follow. Others admit that there are times in which ice or snow will reduce traction to such a point that difficulty is encountered with a trailing axle but, because the percentage of time is so small during which these conditions exist, they do not feel that the additional cost, weight, and maintenance expense of the two-axle drive is warranted. Of interest is the comment of an operator who has to maintain his own roads and found that the use of six-wheelers reduced the necessary upkeep on them.

One manufacturer of trailing axles reports a number of instances of exceptional tire mileage. 60,000 to 80,000 miles on the original rubber is not unusual, and 110,000 and 120,000 miles also have been reported where the jobs were used under more favorable conditions, as with fixed loads to prevent overloading, such as gasoline and milk-tank jobs. An oil company reports a mileage of 62,000 on the four rear wheels of its 40,240 lb. gross weight six-wheeler (6 x $2_c$) equipped with a 633 cu. in. Diesel engine, and the tires are still usable. The front tires were changed at 55,000 miles. The same company has obtained 35,000 miles on a lighter 24,860 lb. gross weight unit.

I have endeavored at various points to show at least some of the theoretical difficulties that tie in with tires and, although they may seem severe, they are not as critical as might be imagined in view of the practical results that have been obtained in the field. The advantages of the six-wheeler in its ability to carry greater gross loads without exceeding the carrying capacity of the tires have relieved this situation somewhat. It must not be forgotten that there also are abuses that

[1] See *Automotive Industries*, Aug. 6, 1927, pp. 198-205; "Six-Wheel Design"; also Aug. 13, 1936, pp. 230-231; "Differential Gear Between Axles in Six-Wheelers", both by P. M. Heldt.

arise in connection with tires on the four-wheeler and that the theoretical ideal seldom prevails in any design. Even if tire costs are slightly greater, the whole picture must be viewed from the ultimate transportation cost, and any increase would be more than outweighed by the savings effected in other ways.

### Axles and Frame Connections

The cross radius rods in the Sterling construction are provided with springs that are supposed to be sufficiently flexible to allow the wheels to track slightly in making a turn. The idea is good, but it is doubted, however, if this flexibility is actually there in view of the otherwise wandering effect that would be imparted to the vehicle. In the Trucktor attachment shown in Fig. 11, the rear ends of the rear springs are permitted a limited amount of lateral displacement in order to attain some tracking.

The use of cross radius rods has been mentioned as a means of relieving the high stress on the bogie trunnions but, of equal importance, is the necessity for maintaining the center and rear axles in parallel relationship, otherwise a condition simulating front-axle toe-in or toe-out effect will cause accelerated tire wear. Where springs are used at the side to tie the axles together, their deflection will vary the bogie wheelbase to a greater extent than will radius rods. This variation indicates the necessity for loading the truck evenly from side to side so that the axles will remain parallel. Spring deflection caused by a road obstacle will produce a similar effect, and normally horizontal radius rods to a lesser extent.

One compensating feature of this construction is that, as the wheelbase increases on the outside due to centrifugal force in going around a turn, the unit is given a tracking effect. This effect seems to be borne out by lessened tire scuffing and easier steering with half-length radius rods over the full-length walking-beam design which keeps the two axles equidistant at all times. The connections of the bogie unit to the frame can be made to allow relative lateral displacement of the axles by placing them, as in the case of radius rods, so as to be inclined to the longitudinal axis of the frame instead of parallel. The use of radius rods is preferable to a spring parallelogram since they permit the spring to function solely as a load-carrying member. The spring ends should be universally mounted to give the axles complete freedom and to relieve the spring leaves of torsional twist.

Propeller-shafts must be long to reduce the universal-joint angularities. Much can be done along this line, particularly with the inter-axle shaft. To secure the maximum length between the joints, the pins must be brought as closely as possible to the pinion shafts. Some joints of the older type do not meet this requirement, and assembly methods must be changed. The Spicer Mfg. Corp. has produced a type in which the end yokes are first assembled on the pinion shafts; the joint proper is then installed by inserting the journal crosses in the yokes by tilting, followed by centralizing them with the bearing units and locking them in with end plates.

One of the standard bogie units having double-reduction

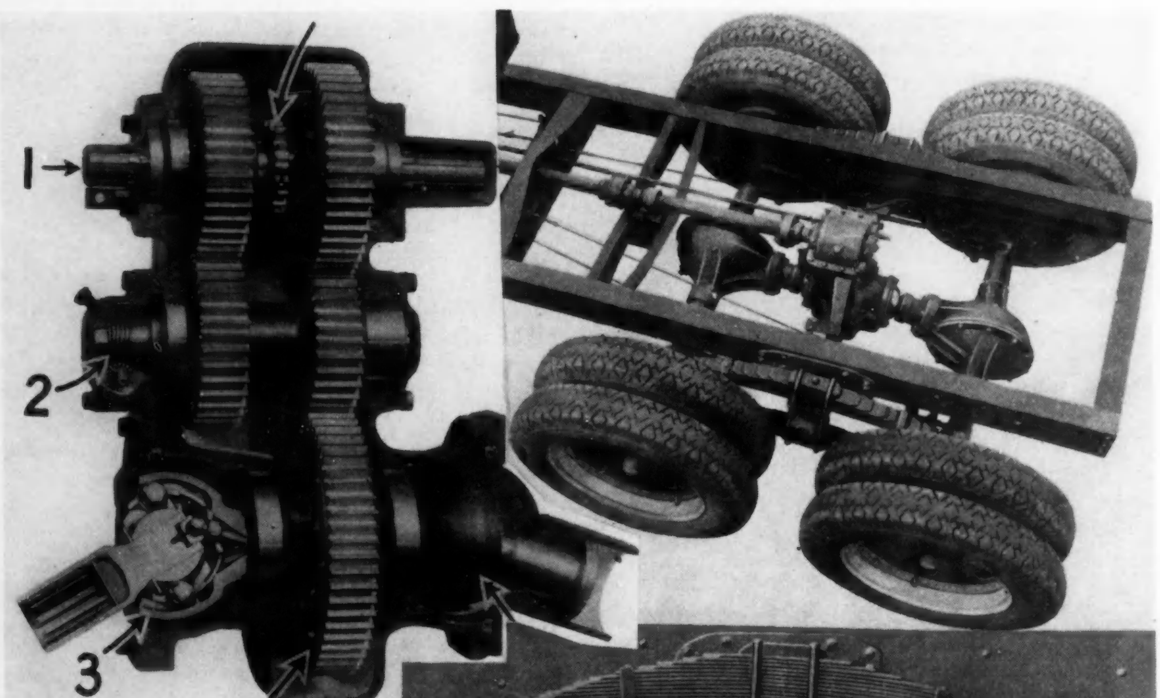



Fig. 12 – Grico Unit with Two-Speed Transfer Case

drive and an inter-axle differential imposes a normal driving angle of 10 deg. on the joint, with a maximum interference angle of 42 deg. Such an angle is ruinous to any joint of the plain-bushing type. For this reason needle bearings are obligatory for this service and owe their present-day popularity to this source.

The axles have a greater rotational movement about the axle-shaft axis than in the case of a four-wheeler, and the universal-joints, radius rods, and suspension system must be laid out in accordance with this fact. Shackles, if used, must be long to allow greater oscillation. They and the spring bolts should be more rugged in view of the more severe duty imposed upon them.

An argument sometimes presented against the two-wheel-drive six-wheeler (6 x 2) is the concentration of all the driving load on a single set of gears and a single pair of axle shafts. Although this concentration is in no way different from the conditions obtained with a tractor-semi-trailer, this fact should be given recognition where the service is more strenuous, and it is to be noted that most manufacturers make a practice of using a heavier driving axle on the "6 x 2" vehicle than in the case of the "6 x 4".

In Germany, Büssing-Nag, V.N.K.A.G., had been in production for a number of years on a six-wheeler (6 x $4_{c+r}$) with two driving axles somewhat similar to the Hendrickson and International arrangements with the differential housings of the axles off-center. A torque tube is incorporated with each axle and anchored in a spherical support on a frame cross-member. The torque tubes are of the same length. The rear-axle tube is supported on a dropped cross-member passing under the center-axle torque tube and thus locating the front end of the rear tube below the center-axle housing.

In the early designs a transfer case at the back of the transmission conveyed the drive to each of the propeller shafts leading to the torque-tube spherical mountings and the universal-joints within. In 1934 a double engine having separate banks of cylinders on a common crankcase drove through separate clutches and transmissions back to the same center- and rear-axle combination. In 1935 two Diesel engines each developing 140 hp. were disposed at opposite ends of the frame as shown in Fig. 15. The gear box of the front engine is located near the center of the chassis and drives the rear axle. The rear engine and its transmission are close coupled and drive the center axle. The engines, drive lines, and axle differential housings are offset as in the previous design. Compressed air is used for gear shifting, and a single-front radiator serves both engines.

A similar type of two-engine drive (6 x $4_{c+r}$) was developed by Ralph Werner as shown in Fig. 16. Each engine drives the adjacent axle. Steam cooling is used, avoiding fan and water-pump power losses. As developed, the gross weight rating of this truck was 50,800 lb., consisting of 11,800 lb. for chassis and cab, 3000 lb. for the body, and 36,000 lb. for the load.

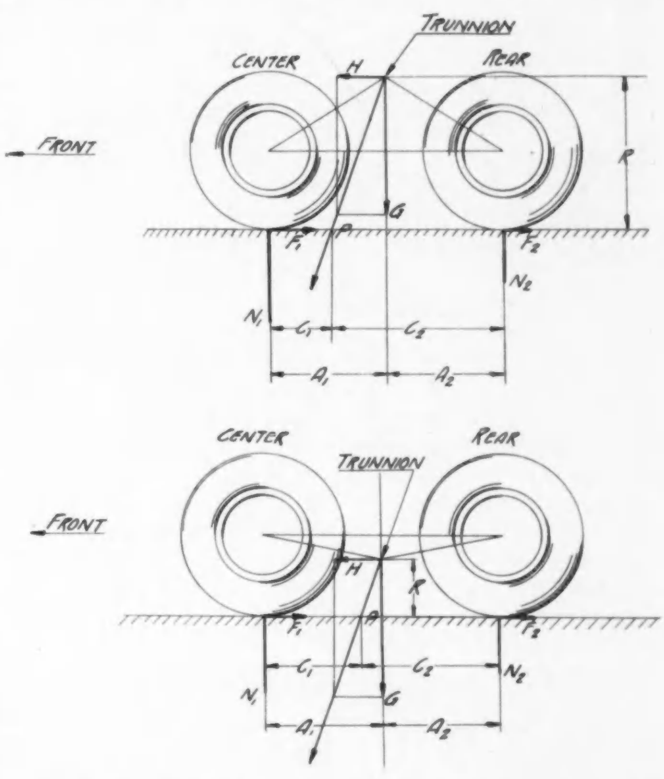
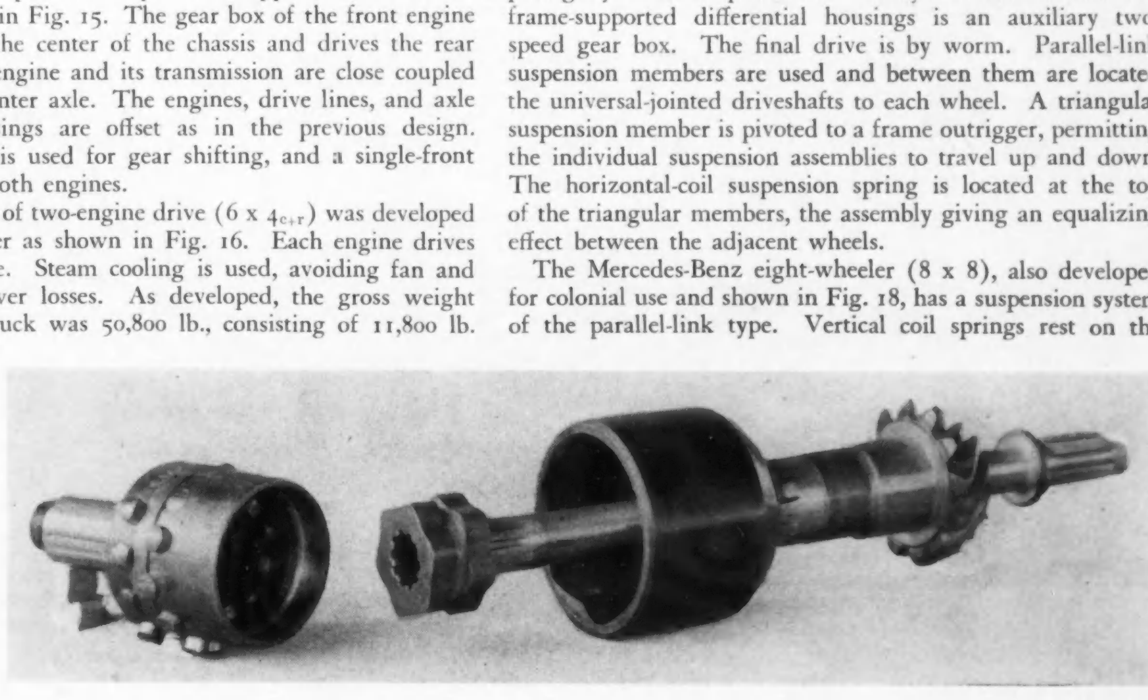



Fig. 13 – Diagram Showing Weight Transfer in Braking

This paper would not be complete without referring to some of the individual wheel-suspension designs developed in Europe. The Krupp cross-country light truck (6 x $4_{c+r}$), shown in Fig. 17, was developed in 1934 and equipped with a 50-hp. Diesel engine having two banks of two horizontal opposing cylinders disposed transversely. Ahead of the two frame-supported differential housings is an auxiliary two-speed gear box. The final drive is by worm. Parallel-link suspension members are used and between them are located the universal-jointed driveshafts to each wheel. A triangular suspension member is pivoted to a frame outrigger, permitting the individual suspension assemblies to travel up and down. The horizontal-coil suspension spring is located at the top of the triangular members, the assembly giving an equalizing effect between the adjacent wheels.

The Mercedes-Benz eight-wheeler (8 x 8), also developed for colonial use and shown in Fig. 18, has a suspension system of the parallel-link type. Vertical coil springs rest on the

Fig. 14 – Mack Cam-Type Power-Divider



upper links and the load is transferred to them from the frame by a normally horizontal equalizer beam. This construction is, of course, applicable to the six-wheeler.

### Light Units in the Operating Field

All operators find that the six-wheeler in its legitimate field is far more economical to use based on a greater number of ton miles than can be obtained from a four-wheeler. Some of the oil companies are experimenting with tank trucks in which a light chassis is used in conjunction with a trailing axle. Certain factors are behind this interest, principally due to the agility of this type of vehicle as compared to heavy units. This agility has been accomplished in some instances by the use of a two-speed driving axle and by the training of the drivers to operate more skillfully by the use of an engine tachometer in which red lines on the dial indicate the desirable engine operating-speed limits. This control prevents hesitation in shifting and it becomes possible for an 18,000 lb. gross unit to negotiate a 4½ per cent grade at 25 m.p.h. or a 9 per cent grade at 9 m.p.h. This truck was using the low rear-axle ratio of 8.57:1 and had a 1934 Ford engine. The high ratio is 6.17:1. There is no doubt that similar technique in the use of a tractor-semi-trailer would improve its performance also. In many instances the operating cost per mile has been reduced to such an extent that a real light-weight six-wheeler movement is now under way in the oil and gasoline distribution field.

On short hauls, where the time required to drain the tanks offsets the running time involved, the lower-priced units have been found to reduce operating costs and make the operation more economical. In some territories where the speed restrictions on vehicles of heavy gross weights are strict and the license fees very high, the smaller type of unit will in a short period of time pay for itself out of its savings of license fees and the additional time gained by overcoming the otherwise speed limitations. Furthermore, in States like Alabama or South Carolina where a 20,000 lb. gross is the maximum weight permitted, the light six-wheeler fits in admirably.

### Experiences of a Large "6 x 6" and "6 x 4" Operator

One large operation that depends little on the use of roads has apparently found scant need for the six-wheeler driving on only the four rear wheels (6 x $4_{c,r}$). There are only a few other places where it fits in, such as an emergency unit for use around flying fields but, for general off-road purposes, drive on all six wheels (6 x 6) has been found desirable. This operation is exceptional and of particular interest in both the 6 x 4 and 6 x 6 types.

Service use has demonstrated that greater consideration must be accorded the question of permissible "lateral float". In a 6 x $4_{c,r}$ vehicle having a gross weight distribution of 11,000 lb. on the rear bogie tires, the tires being 8.25-18 duals with a total capacity of 19,600 lb. for the eight tires, it was found that the life of a set of rear tires was less than 3000 miles, while the front tires were in excellent condition at that mileage. Tests were conducted to determine the reason for the high rate of rear-tire wear, and it was found that it could be traceable only to excessive "lateral scrub" in straight-away operations. The vehicle had a ¾-in. permissible lateral float on each rear axle, this float without distortion of the vehicle springs. When negotiating a turn, the ¾-in. shift of axles promptly occurred, and it was found that they did not return to a normal (tracking) position when the vehicle went into the straight-away. A turn in the direction opposite to the original turn was required to shift the axles back to normal, but too great a turn shifted them to the opposite side of the normal position and they remained there. As long as the axles were in the normal position, trouble was not apparent but, when driving on the straight-away with the axles shifted, a scrub occurred that actually could be heard. Plates were wired to the sides of the springs and the lateral float was decreased gradually until only ¼ in. was allowed on each axle. The axles in shifting on a turn took up the ¼ in. and distorted the springs slightly, but readily shifted back to normal immediately on going into the straight-away, and there was no noticeable effect of scrubbing.

Although it appears a rather general impression that lateral

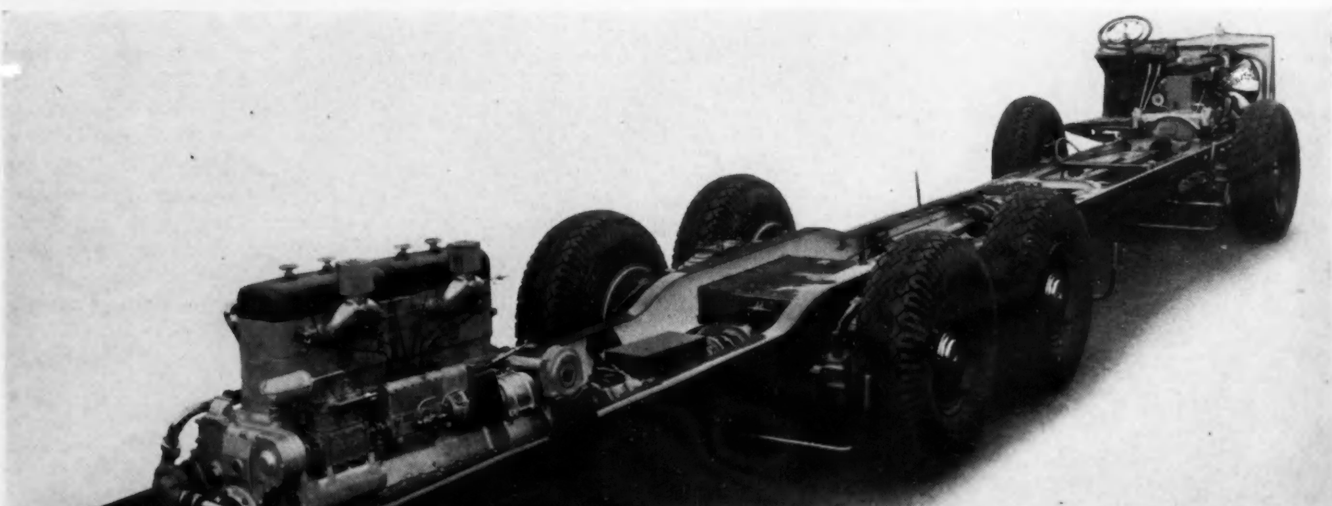


Fig. 15 – Bussing-Nag Bus Chassis with Diesel Engine at Each End

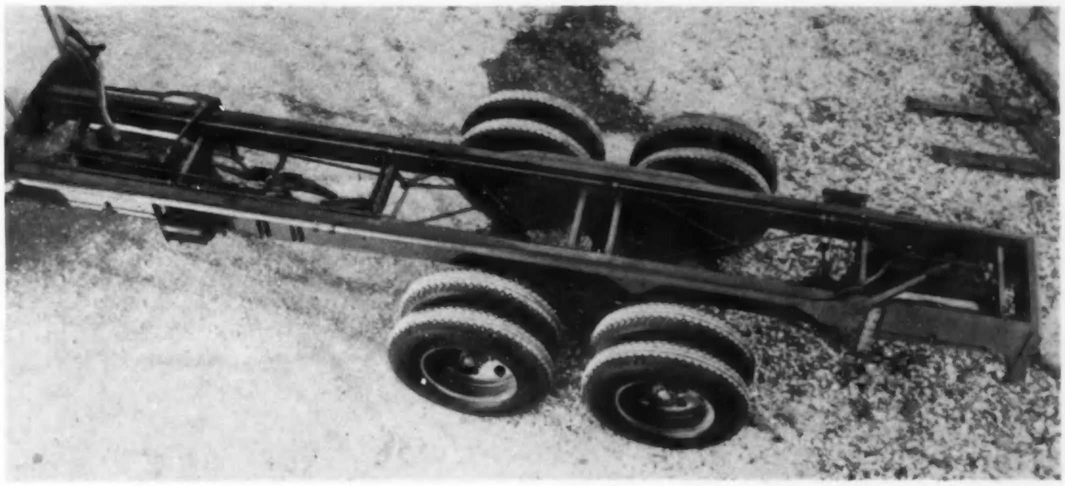


**Fig. 16—Werner Chassis with Center and Rear Axles Each Driven by an Engine**

float must be permitted on the axles when negotiating turns, it is the impression that the float should all be against vehicle-spring distortion and that clearances that permit axle float without spring distortion should be limited to only the amount required to readily assemble the units, not over ¼ in. total per axle. It seems that the probable tire life may be somewhat reduced by so limiting the float, probably 15 per cent less than the life of the same tire were it on a single rear-axle chassis. But, in attempting to alleviate the condition by allowing an appreciable lateral float, lateral scrub is introduced on straight-away operations, possibly reducing the probable life of the tires as much as 60 per cent under aggravated conditions.

In a four-wheel bogie where all four wheels are driven, it has not yet been determined why the axles should shift back to the normal position. Both axles are traveling at the same rate of speed, and every factor tends to offset the slight tendency to shift back to a tracking position. Vehicle-spring side pressure must be resorted to if lateral scrub is to be eliminated, or a differential must be interposed between the two axles. If a differential is added, the rear-most axle can slow up enough to fall back into the tracking position.

The inter-axle differential is not looked upon with favor in this particular off-road service since it is the thought that traction cannot be obtained with differentials that have been developed thus far unless equipped with a manual lock-out mechanism. They claim that sufficient difficulty is experienced already with wheel-slip arising from the action of the axle differential, even when a semi-locking type is employed. In view of trouble experienced with axle-differential pinion gears galling, it is claimed that they will and do gall more readily under center-differential service. In the case of a bogie inter-axle differential, it is the opinion that the advantages obtained do not offset the disadvantages that are introduced, and that one is not needed if the bogie is designed, assembled, and maintained properly. In the case of a center differential between the driving front axle and rear axle (or axles) in a 4 x 4 or 6 x 6 vehicle, there appears to them far more reason why one should be provided.

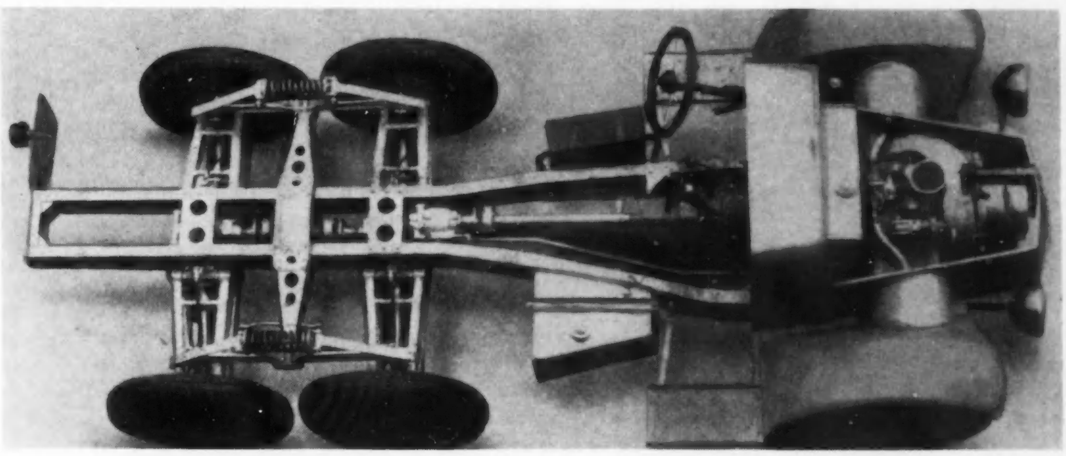


**Fig. 17—Krupp Cross-Country Truck with Individual Wheel Suspension**

A front-axle de-clutching mechanism appears preferable to this operation and costs considerably less than the manual locking mechanism required with a center differential. It has recently been redesigned to eliminate the lower transmission reductions when the front-axle drive is de-clutched. In an eight-forward speed transmission, consisting of a four-speed main transmission and a two-speed auxiliary, the control mechanism is so interlocked that the auxiliary reduction cannot be employed when the front-axle drive is de-clutched. This arrangement prevents the high total reduction usually employed in a 6 x 6 or 4 x 4 vehicle transmission system from being utilized with only the rear axles driving. In a properly designed vehicle having the front axle driven, experience has indicated that a center differential in the auxiliary transmission is not necessary when the vehicle is being operated generally under full-load conditions. But, where a 1/3-2/3 weight distribution is employed, extensive operations with the vehicle

**Table 2 – Estimated Life Cost of Light-Weight Six-Wheelers**

| | | |
|---|---|---|
| | Chassis, $750 | Life, 75,000 Miles |
| | Third Axle, $700 | Life, 150,000 Miles |
| | Two-Speed Axle, $150 | Life, 150,000 Miles |
| | Tank, $1000 | Life, 200,000 Miles |
| 25,000 Miles | Turn in at 25,000 Miles—1 Year | |
| | Maintenance including washing and tires | $ 297 |
| | 50 per cent depreciation on chassis | 375 |
| | 1/6 depreciation on third axle | 116 |
| | Transferring third axle and tank plus painting | 150 |
| | 1/6 depreciation on two-speed axle | 25 |
| | 1/8 depreciation on tank | 125 |
| | | $1088 |
| | Cost of $0.0435 per mile | |
| 50,000 Miles | Turn in at 50,000 Miles—2 Years | |
| | Maintenance including washing and tires | $1122 |
| | 2/3 depreciation on chassis | 500 |
| | 2/6 depreciation on third axle | 232 |
| | Transferring third axle and tank plus painting | 150 |
| | 2/6 depreciation on two-speed axle | 50 |
| | 2/8 depreciation on tank | 250 |
| | | $2304 |
| | Cost of $0.0461 per mile. | |
| 75,000 Miles | Turn in at 75,000 Miles—3 Years | |
| | Maintenance including washing and tires | $2012 |
| | 3/4 depreciation on chassis | 560 |
| | 3/6 depreciation on third axle | 348 |
| | Transferring third axle and tank plus painting | 150 |
| | 3/6 depreciation on two-speed axle | 75 |
| | 3/8 depreciation on tank | 375 |
| | | $3520 |
| | Cost of $0.0469 per mile. | |

unloaded over hard surface terrain has resulted in a high rate of front-tire wear. The front-axle drive-de-clutching mechanism was designed to cope with this condition.

The 6 x 6 construction is applicable to service in the oil fields, lumber camps, quarries, forest-fire fighting, and so on. This type of vehicle introduces considerations that are distinct in themselves, such as the use of torque-dividing differentials between the drives to each end of the vehicle if one is used, and also the question of steering geometry which assumes a different phase than in the 6 x 2 or 6 x 4 vehicle.

### Removable Traction Devices

With the trailing axle, both the Trucktor and Maxi units can be provided with sprockets to form a four-wheel drive. The chains ordinarily are carried in the tool box and, when

**Table 3 – Detailed Basis of Estimated Life Cost**

| | | |
|---|---|---|
| Paint | Yearly | $55.00 |
| Change Oil | Every 1000 miles | |
| (1) Washing truck | Weekly | 2.00 |
| (2) Service, grease, check battery, lights, and tighten | Semi-monthly | 3.60 |
| (3) Repairs | Mileage | |
| Adjust brakes | 4,000 | 2.50 |
| Tune motor | 10,000 | 2.00 |
| Exchange distributor | 10,000 | 2.50 |
| Adjust and oil clutch | 5,000 | 1.00 |
| Exchange water pump | 15,000 | 7.00 |
| Exchange generator | 25,000 | 10.00 |
| Remove carbon | 15,000 | 3.00 |
| Repair wiring and lamps | 35,000 | 10.00 |
| Renew front springs | 20,000 | 7.50 |
| Renew spring shackles | 15,000 | 2.00 |
| Repair radiator and hose | 20,000 | 6.00 |
| Renew steering, knuckle, pins, and bushings | 20,000 | 8.00 |
| Renew spark-plugs | 15,000 | 3.20 |
| Repairs to cab, windshield, and cushions | 25,000 | 10.00 |
| Tank faucets | 35,000 | 20.00 |
| Overhaul steering | 40,000 | 14.00 |
| Overhaul transmission | 35,000 | 25.00 |
| Overhaul rear end | 50,000 | 35.00 |
| Overhaul clutch | 30,000 (Motor Out) | 12.00 |
| Exchange motor | 30,000 | 50.00 |
| Renew universals | 20,000 | 10.00 |
| Reline brakes, true up drums | 25,000 | 20.00 |
| Third axle | 50,000 | 30.00 |
| Tires—(10-Ply) and tubes, each | 25,000 | 28.55 |
| Wheel bearings | 40,000 | 24.00 |
| Repairs to springs | 35,000 | 25.00 |
| Renew battery | 25,000 | 11.00 |

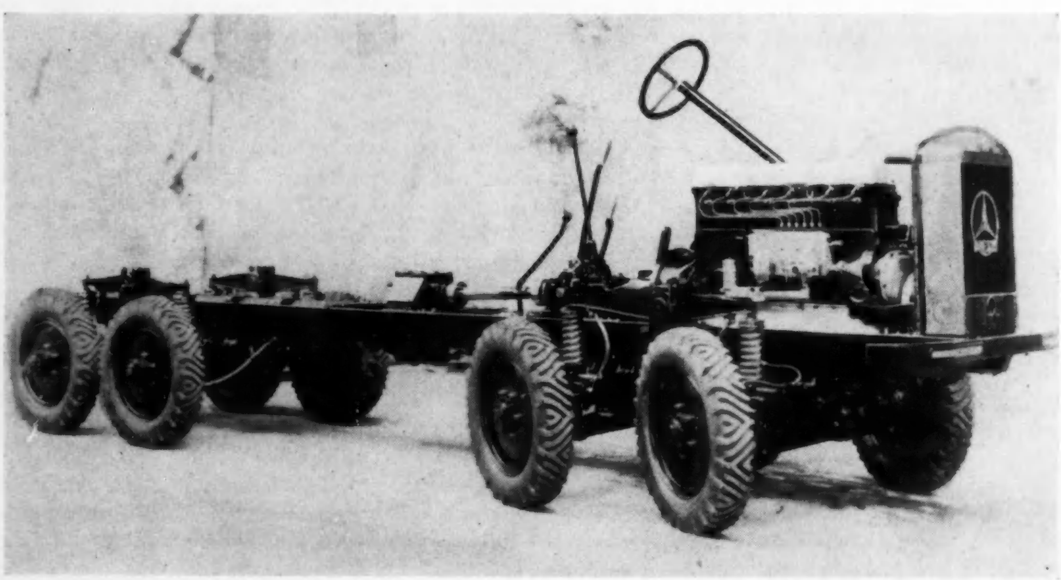


**Fig. 18 – Mercedes-Benz Colonial Truck with Individual Wheel Suspension**

**Table 4 – Data for Profitability Comparison**

Heavy-Duty Four-Wheeler, 1000-Gal. Capacity
Vs.
Light Six-Wheeler, 1000-Gal. Capacity
Operated at Trenton, N. J.

| Unit | Mileage | Total Expense | Fixed | Wages | Operating | Maintenance |
|---|---|---|---|---|---|---|
| Four-Wheeler | 23,965 | $5,455.08 | $816.17 | $1,744.52 | $562.40 | $2,331.99 |
| Cost per Mile | | 22.7¢ | 3.4¢ | 7.3¢ | 2.3¢ | 9.7¢ |
| Figures on above unit collected from 12 months' experience | | | | | | |
| Six-Wheeler | 25,566 | $2,872.53 | $846.52 | $1,174.39 | $468.78 | $382.84 |
| Cost per Mile | | 11.2¢ | 3.3¢ | 4.6¢ | 1.8¢ | 1.5¢ |
| Figures on above unit collected from 9 months' experience | | | | | | |

the road conditions warrant it, the driver can apply the chains to the sprockets located between the dual wheels, which also act as spacers to provide the necessary chain clearance between the tires. This provides an easily convertible two-wheel drive unit to a four-wheel unit. It has been found that ordinary weather conditions only require the extra drive about 10 per cent of the time.

The Hopkins Tractioneer is made up of a series of steel plates connected with clevises and chains which can be applied around the circumference of dual tires thereby giving increased traction on each driving wheel, or it can be applied to the center and rear wheels to form a track for a six-wheeler having either two or four driving wheels. The plates, which extend beyond the center of the duals, are cupped outwardly to give increased traction in soft going. The plates, in swivelling on the chain, lie flat against the ground during contact therewith.

### Light-Weight Truck Operating Costs

In setting up a program to evaluate a maximum cost for the light-weight six-wheeler operations, one oil company estimated its cost as given in Table 2, these charges in turn being based on the data presented in Table 3. From costs spread over a period of twelve months, it is believed that the estimate in Table 2 is substantially correct. Attention is called to the fact that about half the mile rate in Table 2 is depreciation, leaving a maintenance and tire rate of around $0.025 per mile.

This truck is provided with a 1200-gal., four-compartment tank and took the place of a 17,500 lb. gross weight four-wheeler with 1000-gal. tank. The original costs of these two vehicles were $2,900 and $5,000 respectively. The lighter truck easily covers 50 per cent more mileage, or in the neigh-

**Table 6 – Tanker Weight Distribution With Various Loadings**

| | Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|---|
| Unladen | 3005 | 10,675 | 13,680 |

No. 3 compartment with 300 gal. of gasoline and three 5-gal. cans of oil.

| Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|
| 3115 | 12,065 | 15,180 |

Nos. 3 and 4 compartments with 600 gal. of gasoline and six 5-gal. cans of oil.

| Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|
| 3095 | 14,005 | 17,100 |

Nos. 3, 4, and 2 compartments with 900 gal. of gasoline and nine 5-gal. cans of oil.

| Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|
| 3665 | 15,380 | 19,045 |

Nos. 3, 4, 2, and 5 compartments with 1200 gal. of gasoline and twelve 5-gal. cans of oil.

| Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|
| 3360 | 17,695 | 21,055 |

Nos. 3, 4, 2, 5, and 1 compartments with 1500 gal. of gasoline and fifteen 5-gal. cans of oil.

| Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|
| 4160 | 18,645 | 22,805 |

Nos. 3, 4, 2, 5, 1, and 6 compartments with 1800 gal. of gasoline and eighteen 5-gal. cans of oil. This weight is of truck completely loaded.

| Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|
| 3620 | 21,650 | 25,270 |

One compartment of 300 gal. and three 5-gal. cans of oil added at each weighing.

Data on K-47 Dodge, 169-in. wheelbase, 84½-in. C. A., Trucktor axle, 189-in. tank.

**Table 5 – Data for Profitability Comparison**

Heavy-Duty Four-Wheeler, 2000-Gal. Capacity
Vs.
Light Six-Wheeler, 2000-Gal. Capacity
Operated at Fall River, Mass.

| Unit | Mileage | Total Expense | Fixed | Wages | Operating | Maintenance |
|---|---|---|---|---|---|---|
| Four-Wheeler | 44,519 | $10,292.77 | $1,377.33 | $3,648.97 | $1,916.42 | $3,350.05 |
| Cost per Mile | | 23.1¢ | 3.1¢ | 8.2¢ | 4.3¢ | 7.5¢ |
| Figures on the above unit collected from 12 months' experience | | | | | | |
| Six-Wheeler | 11,101 | $1,222.84 | $107.83 | $689.60 | $247.19 | $178.22 |
| Cost per Mile | | 11.0¢ | 1.0¢ | 6.2¢ | 2.2¢ | 1.6¢ |
| Figures on above unit collected from 4 months' experience | | | | | | |

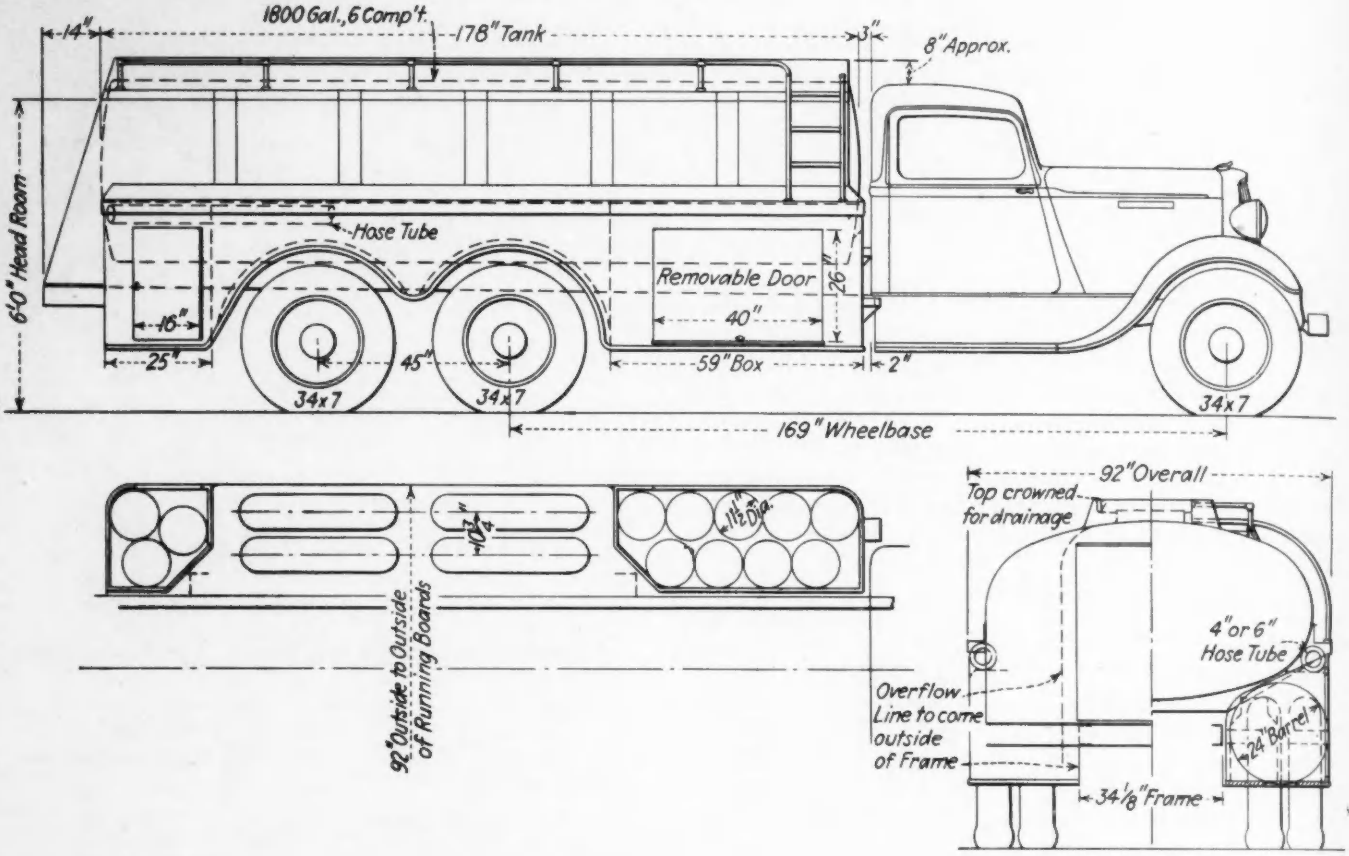



Fig. 19 – Typical Light-Weight Oil Tanker

borhood of 3000 miles per month. The maintenance on the four-wheel truck including tires was $0.050, and the depreciation based on 150,000 miles of life was $0.030.

**Table 7 – Tanker Weight Distribution With Various Loadings**

| | Front, lb. | Rear, lb. | Total, lb. |
|---|---|---|---|
| Unladen | 3500 | 10,990 | 14,490 |
| No. 3 compartment filled and three 5-gal. cans of oil. | 3970 | 12,700 | 16,670 |
| Nos. 3 and 4 compartments loaded and 6 cans of oil. | 3930 | 14,650 | 18,580 |
| Nos. 3, 4, and 2 compartments loaded and 9 cans of oil. | 4820 | 15,880 | 20,700 |
| Nos. 3, 4, 2, and 5 compartments loaded and 12 cans of oil. | 4530 | 18,170 | 22,700 |
| Nos. 3, 4, 2, 5, and 1 compartments loaded and 15 cans of oil. | 5510 | 19,200 | 24,710 |
| All compartments loaded. 18 cans of oil in front boxes. | 5050 | 21,700 | 26,750 |

Data on 704-K White, 169-in. wheelbase, 93-in. C. A., Tructor axle, 189-in. tank.

Table 4 gives a comparison between an 18,000 lb. gross weight six-wheeler with a 1000-gal. tank as against a 17,500 lb. gross weight four-wheeler with a similar capacity tank and operating in a low-wage area and level country. It will be seen that the total expense has been more than halved. Table 5 shows a comparison between two 2000-gal. tankers, the six-wheeler having a chassis of 24,600 lb. gross weight capacity while the four-wheeler is of 30,000 lb. capacity.

## Tanker Equipment

A fully skirted tank truck similar to the six-wheeler designated in Table 5, but with an 1800-gal. tank for gasoline distribution, is shown in Fig. 19. The weights both empty and with various loads, taking in the compartments and front boxes, are given in Table 6 (the 5-gal. cans of oil weigh 68 lb. each).

From the weights given it will be seen that the sequence of unloading the various compartments must be such as to not seriously overload or underload the front axle. It will be noted that the front-axle loading for the fully loaded condition is slightly under the 16 per cent front distribution necessary for steering response over icy roads; yet it is sufficient for wet pavements. However, under critical icy conditions, No. 6 compartment can be left empty in order to attain the desired steering response. A similar weight distribution for a 26,750 lb. gross weight chassis with a six-compartment 1800-gal. tank is shown in Table 7.

The interest of one oil company in investigating the six-

wheel field is indicated by its program in purchasing nine different makes of trucks in order to investigate the full possibilities of the light six-wheeler, and any possible superiority of certain constructions, such as five-speed transmissions, overdrives, and two-speed axles for this purpose.

Transmission ratios are critical and must be considered carefully since the reduction advantage (with small steps) may be overbalanced by the internal friction through the gears. Another oil company is experimenting along the same line with two of its light units with trailing rear axle as represented by data given in Table 8. Performance of these trucks has been found very satisfactory to date.

It can be appreciated readily that this type of service for the lighter trucks is imposing more arduous duty on the powerplant in particular. These efforts undoubtedly will react in improvements which will impart greater reliability and endurance to them. Increased emulsification in the engine oil has been encountered and, in the winter, the strainers must be kept clean to a greater extent. These troubles are overcome in one case by the use of thermostats and a pan under the crankcase. Decreased fuel and oil consumption resulted when going from S. A. E. 40 to S. A. E. 20 oil after reboring the motor. Uniform engine operating temperatures are desirable and, without a pan, it has been found desirable to raise the thermostat control to maintain the water jackets at summer temperatures.

Although the smaller engines can be refitted or replaced economically, this field will decide the more economical method of rejuvenating the engine, be it by means of replaced cylinder sleeves, piston-rings, connecting-rod and other bearings versus unit replacement. The whole situation is still in the experimental stage, and the final results are being anticipated keenly. It is interesting to note that the first trouble encountered in these developments was the basing of the weight distribution on the cargo, resulting in a light front-axle loading which, besides lack of steering response, results in snapping of the shackles due to the prancing of the truck under certain conditions. The error was due to not considering the lighter weight powerplant since the previous six-wheelers used were of the heavy-duty type with heavy engines which in themselves added considerably to the front-axle loading. However, appraising conditions in their true light, the present design has overcome these early difficulties. One

### Table 8 – Light-Weight Six-Wheeler Data

| | |
|---|---|
| **Truck A:** | |
| Tank Capacity | 1,217 gal. |
| Oil Capacity | Fourteen 5 gal. cans |
| Total Length | 25 ft., 6 in. |
| Gear Ratio | 6.6:1 |
| Engine not governed, top operating speed 40 m.p.h. | |
| Tire Size | 32 x 6 all around |
| Empty Weights— | |
| Front Axle | 3,065 lb. |
| Both Rear Axles | 7,420 lb. |
| Tructor Axle Only | 3,620 lb. |
| Total Weight | 10,210 lb. |
| Loaded Weights— | |
| Front Axle | 4,060 lb. |
| Both Rear Axles | 14,910 lb. |
| Tructor Axle Only | 7,450 lb. |
| Total Weight | 18,615 lb. |
| **Truck B:** | |
| Tank Capacity | 920 gal. |
| Oil Capacity | Ten 5-gal. cans |
| Total Length | 21 ft., 9 in. |
| Gear Ratio | 6.6:1 |
| Engine not governed, top operating speed 40 m.p.h. | |
| Tire Size | 32 x 6 all around |
| Empty Weights— | |
| Front Axle | 2,435 lb. |
| Both Rear Axles | 7,625 lb. |
| Tructor Axle Only | 3,750 lb. |
| *Total Weight | 10,210 lb. |
| Loaded Weights— | |
| Front Axle | 3,330 lb. |
| Both Rear Axles | 13,370 lb. |
| Tructor Axle Only | 6,955 lb. |
| Total Weight | 16,850 lb. |

*The total empty weights of these two trucks happened to be the same although one is of 1217-gal. capacity and the other of 920-gal. capacity. This condition is due to the fact that the 1200-gal. unit has a new light-weight steel tank, and the 900-gal. unit has an old-style heavy steel tank.

Maintenance:

| Truck No. | Installation Date | Total Repairs | Total Tires | Total Miles |
|---|---|---|---|---|
| A | 7/6/35 | $41.52 | $2.50 | 8769 |
| B | 9/4/35 | 12.83 | | 3313 |

Gasoline and Oil Consumption:

| Truck No. | Miles per Gallon of Gasoline | Miles per Gallon of Oil |
|---|---|---|
| A | 6.6 | 601 |
| B | 5.9 | 571 |

### Table 9 – Comparative Cost Data, Heavy-Duty Trucks

| Four-Wheel Truck ($4 \times 2_r$) | | Six-Wheel Truck ($6 \times 4_{c+r}$) | |
|---|---|---|---|
| Registered Capacity | 17,500 lb. | Registered Capacity | 22,000 lb. |
| Light Weight | 14,500 lb. | Light Weight | 18,000 lb. |
| Gross Weight | 32,000 lb. | Gross Weight | 40,000 lb. |
| (Semi-Van Body) | | (Semi-Van Body) | |

| | Four-Wheel Truck | Six-Wheel Truck |
|---|---|---|
| Gasoline Consumption | 11,938 gal.; cost, $1350.44 | 14,980 gal.; cost, $1679.99 |
| Oil Consumption | 551 qt.; cost, 79.00 | 375 qt.; cost, 46.86 |
| Shop Repair Work | 216.72 | 247.52 |
| Garage Maintenance Overhead (apportioned) | 647.50 | 814.00 |
| Insurance, Registration, and Taxes | 637.00 | 800.80 |
| Depreciation on Chassis and Body | 1602.77 | 2095.23 |
| Tire Depreciation on Mileage Basis | 482.46 | 888.39 |
| Total Costs | $5015.89 | $6572.79 |
| Mileage Traveled | 44,672 | 49,355 |
| Average Working Days in Year | 300 | 300 |
| Total Expense per Mile | $0.112 | $0.133 |

(Above figures are based on 1935 operations)

remaining factor is that the smaller engines do not give the braking of the larger ones and greater reliance must be placed on the wheel brakes. However, tests made in the low ratio of a two-speed axle equipped 18,800 lb. gross weight truck showed that in direct drive it could be held at 30 m.p.h. on a 4 per cent grade over ½ mile without touching the brakes. Similarly on an 8 per cent grade in the low gear of the transmission, it was held at 18 m.p.h. for ¾ mile.

I do not wish to be accused of condoning the under-powering of a vehicle, be it a six-wheeler, a tractor-semi-trailer, or any other type. Each type has its usefulness if kept within a sane power-weight ratio. One factor in favor of the oil tankers is the fixed load that can be taken on and the diminishing load; it is possible that these experiences may not be applicable to other transportation problems. It is admitted that engine life, for instance, is not as great as with a larger engine, but today's unit-replacement system presents a different aspect in the economic set-up.

### Heavy-Duty Truck Operating Costs

A heavy-duty operation will be noted in Table 9 where a 32,000 lb. gross weight four-wheeler is compared with the same model chassis but of six-wheel design with an all-drive bogie unit. Semi-van bodies are used in this service, and it is to be noted how small is the added operating cost per mile of the six-wheeler. On the ton-mile basis, the six-wheeler is the more economical for, by an increased expenditure of 19 per cent, a 26 per cent greater load can be transported. In comparing the costs in Table 9 with those of the previous tables it must be remembered that the type of service and territory traversed makes all the difference in the world and comparisons between different operations are not fair unless all conditions are identical.

One oil company found that its 40,000 lb. gross weight six-wheeler operating in and around Baltimore cost $0.320 per mile where the average speed was only 10 m.p.h. and the highest speed 26 m.p.h. Transferring this same truck to make a 460 mile round trip a day in North Carolina in level territory, the cost on a mile basis was cut to $0.144. The insurance has a slight influence since it was halved. A 34 m.p.h. average speed was obtained with a top speed of 38 m.p.h. The number of miles traversed in open-country work is a big factor since the average fixed costs run from 35 to 40 per cent of the total and the wages 20 per cent. Any open-country truck which is transferred to city work will immediately cause maintenance cost to go up about 35 per cent due to the increased use of the clutch and shifting of gears with the resultant wear and tear on the driving mechanism. To mitigate against such abuse, the city-driven truck should be provided with a larger engine in order to relieve the drive lines and axles of the stress incident with the use of a small engine. The larger engine permits a smoother pick-up and prevents the "racing-through" type of driving that invariably results with a small engine.

### Conclusion

In spite of all the arguments over the details of design and operation of the six-wheeler, it has carved its niche in the transportation field and, regardless of any controversies that might arise, it has proved itself to be not only a dependable vehicle capable of coping with problems where the four-wheeler could not be applied, but it has demonstrated in every legitimate application that it is the more economical vehicle of the two on a ton-mileage basis. The design of any mechanism can be applauded when it stands on its own feet economically. There are minor points that undoubtedly will be improved as in the case of any machine. It would be a sad lot for truck transportation had the six-wheeler never been developed and reached its present high point of performance.

Further information on the development of six-wheel trucks with other data is given in an earlier paper[2] by the author.

[2] See S.A.E. Transactions, October, 1932, pp. 387-402; "Six-Wheel Trucks", by Austin M. Wolf.

## Discussion

# Attachments May Lower Factors of Safety

—D. C. Fenner
*Mack-International Motor Truck Co.*

To be safe a structure of any description, whether it be a stationary one like a bridge or a building or a moving one like a vehicle, must be designed with margins of surplus strength and resistance to strain and shock. So vital is it that these margins be provided that they are universally termed factors of safety.

Nowhere are factors of safety of greater importance than in heavy, moving vehicles carrying persons and property over the public highways not only because the shocks and strains are so greatly magnified by speed but also because the safety of all vehicles on the road is dependent to such an extent upon the safety of each.

To be truly safe a vehicle needs factors of safety in its performance ability and its braking ability as well as in its structural capacity. It is, therefore, a serious matter to threaten these factors of safety by alterations and additions made as afterthoughts outside the factory of the original producer and often done without reference to the balance between capacity, load distribution, ability, and braking originally provided.

One of the first attempts to combat this menace was to constitute the Motor Vehicle Commissioner of Connecticut as the sole judge of the capacity of an orphan vehicle.

Next in line came Pennsylvania to establish a definite relationship between chassis weight and gross vehicle weight.

Next, some of the largest fleet operators adopted their own ratio between net chassis weight and gross vehicle weight.

Some years later, California recommended, and Rhode Island adopted, a performance requirement of 20 m.p.h. on a 4 per cent grade, corresponding to a performance factor of 100.

The latest attempt to limit gross vehicle weight – also from California – establishes a ratio between it and the tare weight of vehicle, ready to go to work.

From many States we have repeated suggestions that gross vehicle weight should be restricted to conform to the existing limitations of tire, brake, engine, and chassis capacity of the vehicle in question.

It has even been suggested that we combine all of these proposals, calculate the gross vehicle weight of a particular unit or combination on each of the four bases proposed, namely:

(1) Tire capacity, for distribution
(2) Brake capacity, for deceleration
(3) Engine capacity, for acceleration
(4) Chassis capacity, for structural strength.

Having calculated all four, we then adopt or assign as the official gross vehicle weight the lowest of the four values.

We can then subtract the actual tare weight of the vehicle, ready to go to work, from this value and the remainder becomes the official payload-capacity rating.

Certain it seems that either one of two things is bound to happen:

Either the producers and users of motor vehicles must see to it that adequate factors of safety are maintained – that the vehicles on the road are structurally strong enough to support the loads they carry, powerful enough to keep their place in traffic without being an obstruction to other vehicles, and capable of being brought to a stop within a safe distance under all conditions – or the politicians in panic at the rising toll of highway accidents will clamp down and clamp down hard. It is for automotive engineers to lead the way in establishing sound principles for the guidance of both users and regulating authorities.